CONDENSATION HEAT TRANSFER STUDY FOR POLAR PROFILE TUBES AND THEIR COST OPTIMISATION

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CERTIFICATE

This is to certify that the work presented in this thesis entitled 'Condensation Heat Transfer Study for Polar Profile Tubes and their Cost Optimisation' by Mr. N.K. Patnaik has been carried out under my supervision and has not been submitted elsewhere for a degree.

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NK Patnaik.

N.K. Patnaik

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NOMENCLATURE

Symbols	Description	Unit
A_{cn}	Area of cross section of tubes	m^2
AF	Area factor	dimensionless
A_i	Inner Surface area of tubes	m^2
A_o	Outer surface area of tubes	m^2
ASEC	Annual Saving in energy	Rs.
CE	Cost of energy	Rs.
CEN	Cost of energy	Rs./kW.hr.
CEPU	Cost of production of energy	Rs./kW.hr.
C_{mtl}	Cost of material	Rs./kg.
C_p	Specific heat	J/kg. ^oC
p	Pressure drop	N/m^2
T	Temperature drop	^oC
d_i	Inner diameter of tubes	m
dm	Differential mass flow rate	kg./s.
d_o	Outer diameter of tubes	m
E_p	Pumping energy	J
f	Friction coefficient	dimensionless
FC	Fixed cost	Rs.
g	Acceleration due to gravity	m/s^2
h_{fg}	Latent heat of vapourisation	J/kg.

Symbols	Description	Unit
h'_{fg}	Modified latent heat of vapourisation	J/kg.
HMD	Hydraulic mean diameter	m
h_s	Steam-side (local)heat transfer coeff.	$W/m^2\ {}^oC$
$\bar{h}_s$	Steam-side(Average) heat transfer Coeff.	$W/m^2\ {}^oC$
HTCF	Heat transfer coefficient factor	dimensionless
IR	Interest Rate	%
Ja	Jakob No., $C_p \Delta T/h_{fg}$	dimensionless
k	Thermal conductivity	$W/m\ {}^oC$
L	Length	m
LMC	Loss of material cost	Rs.
LMTD	log mean temperature difference	${}^oC$
M_m	Mass of tube material	kg
n	Average No. of tubes in the condenser in one vertical row	dimensionless
Nu	Nusselt No.	dimensionless
p	Pressure	N/m^2
PER	Perimeter	m
PLMC	Percentage loss in material cost	%
Pr	Prandtl No.	dimensionless
PSEC	Percentage Saving in energy cost	%
PSEEC	Percentage saving in electrical energy consumption.	%
PSTC	Percentage saving in total cost	%

Symbols	Description	Unit
PW	Present worth	Rs.
P_w	Pumping power with one set of two tubes in the condenser	W
PWCE	Present worth of energy cost	Rs.
PWF	Present worth factor	dimensionless
$\dot{Q}$	Rate of heat transfer in one set of two tubes in the condenser.	W
$\dot{Q}_g$	Rate of heat generation per unit volume	W/m^3
$\dot{Q}_T$	Rate of heat transfer in the complete condensing unit.	W
R	Radius of circular profile	m
R_c	Radius of curvature	m
Re	Reynold's No.	dimensionless
R_m	Miscellenous thermal resistance	$m^2\ ^oC/W$
RMF	Radius multiplication factor	dimensionless
R_s	Steam-side thermal resistance	$m^2\ ^oC/W$
R_w	Water-side thermal resistance	$m^2\ ^oC/W$
SECT	Total saving in energy cost	Rs.
SEECY	Saving in electrical energy consumption per year	kW.hr.
S_P	Semi-perimeter	m
STCT	Total saving in total cost	Rs.
STCY	Saving in total cost per year	Rs.
t	Thickness of tubes	m
T	Temperature	^oC

Symbols	Description	Unit
TC	Total cost	Rs.
TC_m	Total cost of material	Rs.
TCE_p	Total cost of energy for pumping	Rs.
TP_w	Total pumping power for the complete condensing unit	W
$T_{sat.}$	Saturation temperature	^oC
T_w	Wall-Surface temperature	^oC
$T_{ref.}$	Reference temperature	^oC
u	Velocity in the 'x' direction of the condensate	m/s
u_m	Mean velocity of the condensate	m/s.
U_o	Overall heat transfer coefficient, based on outer surface area.	$W/m^2\ ^oC$
V^o	Velocity	m/s.
$\dot{\forall}$	Volume flow rate of water for one set of two tubes in the condenser	m^3/s.
$\underline{V}$	Velocity vector	m/s.
$\dot{\forall}_T$	Total volume flow rate for the complete condenser	m^3/s.
x	Direction along the periphery of the tube	m
$\underline{X}$	Body force vector in 'x'direction	N
y	Direction perpendicular to the tube surface	m

Greek Symbols	Description	Unit
β	Angle	Radian
δ	Boundary layer thickness	m
$\underline{\nabla}$	Vector gradient operator	m^{-1}
∇^2	Laplacian operator	m^{-2}
$\underline{\chi}$	Body force vector	N
μ	Dynamic viscosity	$N.s/m^2$
ϕ	Viscous dissipation of energy	s^{-2}
ρ	Mass density	$kg./m^3$
θ	Angle	Radian

Subcripts	Description
c	circular
e	existing
l	liquid condensate
op	optimum
p	polar
w	water for cooling

Abbreviations	Description
I.G.L.	Interactive Graphics Library
P.I.R.C.	Progressively Increasing Radius of curvature.

ABSTRACT

Condenser is one of the most important components in all thermal power plants, large refrigeration and air-conditioning units and in many of the chemical and process industries. Hence, the use of this equipment embraces quite a large arena of the industrial systems. The cost of the condenser, therefore, forms a vital part of the total cost of the above systems. Any effort to bring about a reduction in this cost is an attractive and useful proposition. This thesis is devoted to a study in this direction.

Most of the condensers for commercial purposes, use circular tubes for carrying cooling water through them, which causes condensation over the tubes. If the tube-profiles are changed in a manner, such that, they are drawn in the direction of gravity, the effective gravity force, helping in draining out the condensate from the tube surfaces will be larger than that of the corresponding circular profile tubes. Moreover, by suitably adjusting the radii of curvature, so that they increase continuously from the upper generatrix to the lower generatrix, a favourable pressure gradient is made to develop, which also helps in a faster removal of the condensate from the surface of the tubes. Hence, a thinner film of condensate is left over the tubes having less thermal resistance and as a

consequence, the heat transfer rate is increased. For a given condenser heat-duty, therefore, less surface area is required and, thus, there is a reduction in the first cost of the equipment. Such tube profiles are named as 'progressively increasing radius of curvature' (P.I.R.C.) profiles.

This concept has been explored in the present work. Two tube-profiles, given by the polar curves '$r=a\theta$' and '$r=ae^{\theta}$', reflected symmetrically about their vertical axes, have been used in this context. The effect of using these profiles on the augmentation of heat transfer in the condenser in comparison with the corresponding circular profiles is also studied.

The first cost of the condenser, however, is not representative of the complete picture of its cost. The operating or running cost of the equipment should also be considered in addition to its first cost. The operating cost of a condenser is due to the energy, consumed in the pump for circulating cooling water through the tubes. A complete cost analysis is also carried out in this thesis to obtain the optimum condenser tube dimensions on the basis of minimum total cost of the equipment. A real-life problem is taken by considering the condensing unit of the 110 MW unit of 'The Panki Thermal Power Station, Kanpur', in order to perform this cost analysis. The reduction in size of this condenser and the consequent decrease

in its first cost have also been determined, when circular profile tubes of this condenser are replaced by the already mentioned P.I.R.C. profile-tubes.

Since, the solution of the above problem becomes intractable by pure analytical methods, numerical techniques with the help of a digital computer (DEC-1090) have been employed for the solution. Well-documented computer programmes in FORTRAN language have been developed for this purpose. Necessary graphs and the shapes of the P.I.R.C. profiles for the condenser tubes are drawn by using the subroutines from 'IGL' (Interactive Graphics Library).

CHAPTER-I

1.1 INTRODUCTION :

Condensation occurs when the temperature of a vapour is reduced below its saturation temperature. In industrial equipments, the process commonly results from contact between the vapour and a cool surface. The latent energy of the vapour is released, heat is transferred to the surface and the condensate is formed. Another common mode of condensation is 'homogeneous condensation', where vapour condenses out as droplets, suspended in a gas phase to form fog. 'Direct condensation' occurs when vapour is brought in contact with a cold liquid. Among the modes of condensation, surface condensation is the one, which is most frequently encountered and hence forms the matter of most concern for discussion and study.

Out of the two different categories of SURFACE CONDENSATION, the one, called 'dropwise-condensation' has aroused more curiosity in the scientific community. In this type of condensation, the vapour impinges on the cool wall, decreasing its energy and thereby liquefying and forming drops of condensate. These drops grow by direct condensation of vapour on them and by coalescence with neighbouring droplets, until they are swept off the surface by the action of gravity or other body forces, surface-tension, and/or shear

stresses due to vapour flow. As the drops move, they coalesce with other droplets in their path, sweeping a portion of the surface clean so that condensation can begin anew. The details of 'dropwise condensation' are not completely understood, but it is known to take place under circumstances where the liquid condensate does not wet the surface. The reported coefficients of heat transfer in 'dropwise condensation' are found to be an order of magnitude (10 times) higher than 'filmwise condensation', to be discussed later in this chapter. To achieve 'dropwise condensation' on condensing surfaces, the surfaces have to be coated. Silicones, cupric oleate, several organic polymers and an assortment of waxes and fatty acids are often used for this purpose. However, such coatings gradually lose their effectiveness due to oxidation, fouling or outright removal and film condensation eventually occurs. Permanent coatings of noble metals (gold, silver, rhodium, palladium, platinum) or teflon overcome this difficulty. The substantial increase in cost, however, far outweighs the advantages, gained there from. Although, 'dropwise condensation' of steam has been studied for over fifty years, this unique area of heat transfer has remained as a laboratory curiosity, primarily because permanent hydrophobic coatings have not been developed to the satisfaction of condenser designers. In a very recent

paper [20], it has been presented that organic coatings are successful in promoting good quality 'dropwise condensation' for periods upto around 12 000 hours. For a continuously running powerplant condenser, this requires a change of coating, roughly, after every 1.5 years. This is an attractive proposition. But the uncertainity in this prediction by the authors has been reflected in their pronouncement, "The possible deterioration of the heat transfer with exposure time to steam remains to be determined." So this sends us back to the original impasse. If, some day, the technique of 'dropwise-condensation' is achieved to the level of perfection, the dream of 'steam cars' will become a reality. At present, however, it seems to be a little distant reality.

Since, it has been difficult to sustain 'dropwise condensation' commercially for long periods of time, conservative designs are based on operation with the other significant mode of 'surface condensation,' called the 'filmwise condensation.' In this mode of condensation, the liquid condensate forms a continuous film, which covers the surface and takes place when the liquid wets the surface. This film flows over the surface under the action of gravity or other body forces, surface-tension and/or shear stresses due to vapour flow. Heat transfer to the solid surface takes place through the film, which forms the greatest part of thermal resistance.

Condensation heat transfer can be studied from either the macroscopic or microscopic point of view, as is the case in anyother physical phenomenon. The ultimate objective of the engineer is to understand the phenomenon so that it can be described precisely and hence, its behaviour can be predicted for design purposes. The macroscopic view point considers matter as continuum and requires the introduction of certain phenomenological laws, the macroscopic transport equations, in addition to the conservation equations of mass, momentum and energy. On a microscopic level, dealing with molecules, atoms and subatomic particles, only the conservation principles need be considered. However, because of limitations in describing these particles and in relating the large number of particles, which constitute a reasonable size system, one resorts to the concept of statistical mechanics. For practical applications to engineering systems, however, only the macroscopic method of study is of relevance.

Most of the extensive research in the field of condensation heat transfer has been devoted to the understanding of the basic process. In recent years, however, there has been an increased emphasis on techniques to augment this two-phase heat transfer process. Efforts are continuing to devise means for predicting the heat transfer coefficient for condensation more accurately and to increase the heat transfer coefficient. This will result in reduced condenser sizes and, hence, the first cost of the unit will be reduced.

The increase in heat transfer coefficient will result, if 'dropwise condensation' can be sustained or if the thickness of liquid film, in case of 'film condensation', can be reduced. The first of these alternatives is of no practical interest, at least for the present, in the light of what has been already discussed on the topic of 'dropwise condensation' in this section. Hence, almost all practical methods, used to augment condensation heat transfer, aim at achieving the second alternative i.e. reducing the thickness of the condensate film on the condensing surface. In most of the industrial applications, e.g. large thermal power plants and refrigeration units, condensation takes place in a closed shell over tubes, that carry the coolant through them. Fig. (1.1) shows the outline of such an arrangement, as used at the 'PANKI THERMAL POWER STATION, KANPUR.' Such a process of condensation is termed in technical jargon as 'vapour space condensation' or 'external condensation.' The application of condensation inside tubes, called 'In-tube condensation,' enjoys very limited applications like domestic refrigerators and is, therefore, not discussed further.

The above discussions lead one to conclude that from the engineering viewpoint, the study of 'film condensation' by macroscopic modelling, forms the most important basis for studying the phenomenon of condensation.

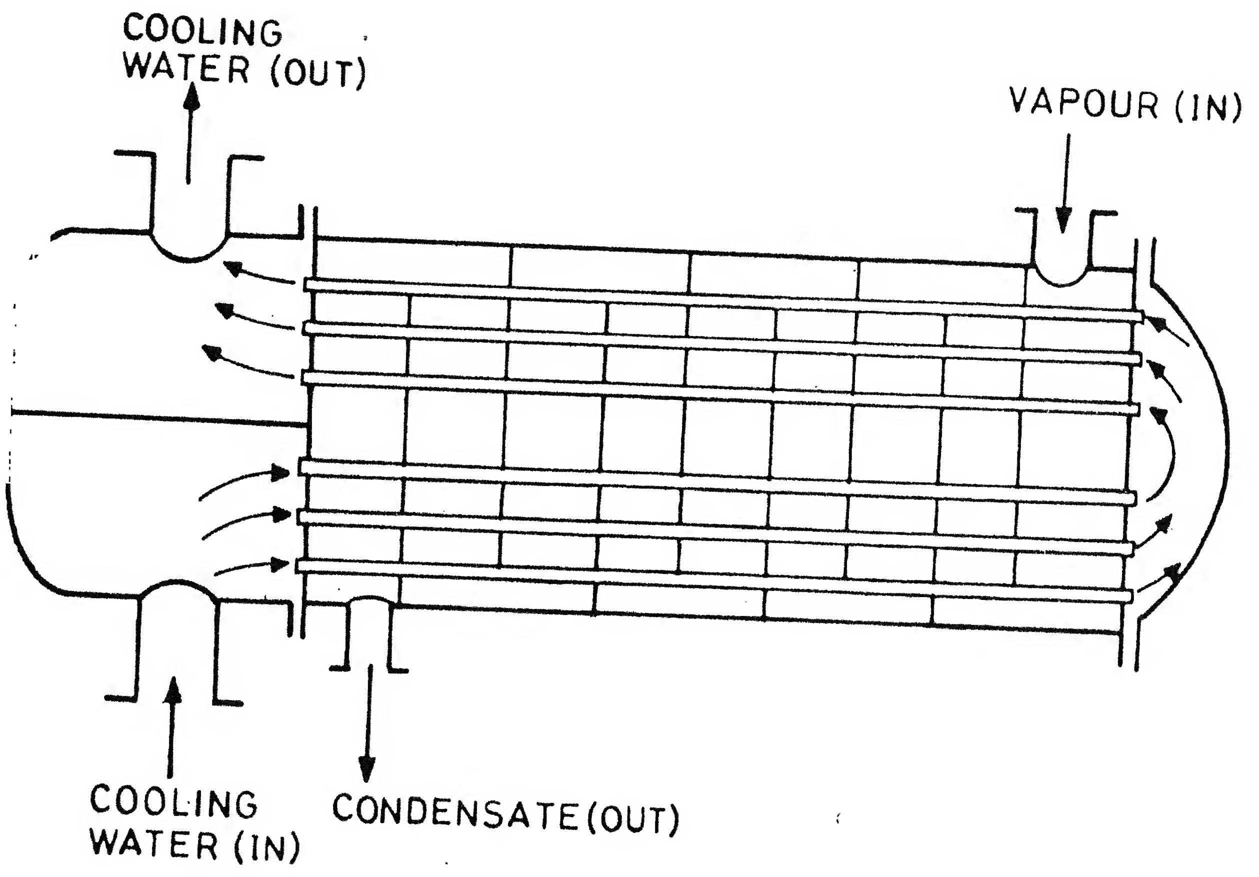


FIG. 1.1 OUTLINE OF THE SHELL AND TUBE ARRANGEMENT IN THE CONDENSER.

1.2 REVIEW OF PREVIOUS WORK :

Most of the condensers, used in commercial practice are designed on the basis of 'filmwise condensation.' The liquid film, formed over the condensing surface, forms a significant source of thermal resistance in the path of heat transfer from the vapour-to-coolant heat exchange process. Any attempt to reduce the thickness of this liquid film by augmenting the condensate drain-out rate, formed over the condenser tubes will, therefore, reduce this thermal resistance and, will, thus, enhance the heat transfer coefficient of the vapour-side.

Much work has been done in this field, dealing with the augmentation of film condensation process and many research papers have been published over the years. A.E. Bergles [14,15] has made a very useful review of most of this ever-widening literature. He has dealt with these techniques of augmenting condensation heat transfer under the categories of (i) active techniques and (ii) passive techniques. The active methods are those which require external power to effectuate the process such as the use of mechanical aids, surface vibration, fluid vibration, electrostatic fields, 'injection and suction' etc. Passive methods, on the otherhand, require no such external power. Some of the passive techniques, used successfully include treated surfaces, rough surfaces, extended surfaces, displaced enhancement, swirl-flow devices,

surface-tension devices and liquid additives. From the point of view of what has been done in the present thesis, the development of surface-tension devices as a passive technique, needs a little elaboration.

Gregorig [15,16] was the first to point out the possible advantages of using surface-tension-driven-forces in enhancing the condensing heat transfer coefficient. His concept of transforming this idea into practice materialized in the development of what is known as the 'fluted tube', as shown in Fig. (1.2). This is essentially a tube with fine corrugations on the surface. The curvature of the surface of the condensate film, formed on a fluted tube is such that the condensate is driven by surface-tension-induced forces from the ridge to the trough. The mechanism, responsible here is that the curvature of the film-surface on the ridge side gives an increased pressure there, while the pressure on the trough side is reduced by reversing the curvature. Hence, there is a pressure gradient from the ridge to the trough, which helps in driving the condensate from the ridge. This leaves a very thin film over the ridge with low thermal resistance. Thisassists in the augmentation of heat transfer. Since, the 'fluted tube' is very complex in shape, incorporation of this device in heat exchanger tubes would require a detailed a-priori economic analysis.

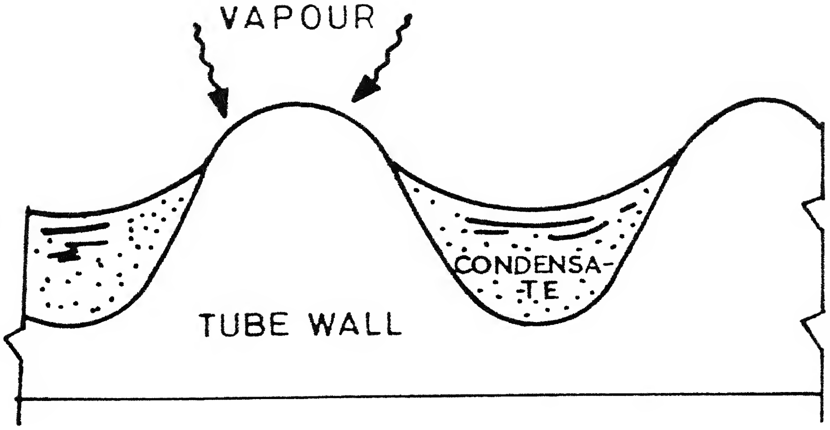


FIG. 1.2 PROFILE OF THE CONDENSING SURFACE DEVELOPED BY GREGORIG.

A relatively simple concept for augmenting condensation heat transfer has been pronounced by G.G.Shklover et. al. [1]. In this paper, the authors have conceptualised the use of two very important factors, which help to enhance the condensate drain-out rate from the condensing surface. Circular profile tubes are moulded into the shape, as shown in Fig. (2.1.b), keeping the perimeters constant. Since, the tubes are drawn more in the direction of gravity in comparison to the circular ones, the contribution of gravity force will increase the drain -out rate of the condensate. Moreover, if in the changed tube profile, the radius of curvature is made to increase continuously from the upper generatrix to the lower generatrix, a favourable surface-tension pressure gradient develops, which assists in the further enhancement of the liquid drain-out rate. Both these effects viz. (a) Gravity force and (b) surface-tension force, in unison, help in the augmentation of condensation heat transfer. In spite of the concept being excellent, the paper does not incorporate a clear and complete solution of the problem. However, a suggestion has been given by the authors that application of this concept to real-life problems, such as condensers in steam-power-plants, will be quite useful. The profile of the tubes, chosen by the authors is that of a logarithmic spiral, which is a little complex, the polar form of which is given by '$r=ae^{\theta\cot\alpha}$'.

No review of literature in the area of film-condensation will be complete without mentioning the pioneering paper of Nusselt [17], which was published in the year 1916 and still remains the only original paper to describe the phenomenon of film-condensation analytically. Improvements have been made over the years, but excepting the behaviour of liquid metals, the original theory has been reasonably successful. Even the authors in [1] have used the same analytical approach as was used by Nusselt [17] to formulate their problem.

1.3 PRESENT WORK :

The present work is mostly an extension of the paper [1]. A detailed and complete solution is obtained for the problem. The suggestion of the authors in [1] is also incorporated. The complete analysis is made on the basis of data, collected from the 'PANKI THERMAL POWER STATION, KANPUR', for the condenser of their 110 MW. unit.

Two tube profiles, represented by the polar curves '$r=a\theta$' and '$r=ae^{\theta}$', respectively, are considered to take advantage of the augmenting effects, mentioned in Section (1.2). The economics of this change is also discussed.

Employment of the above mentioned tube profiles, however, is against the conventional practice of using circular tubes in condensers. Whenever, a change is

contemplated in the conventional practice, more often than not, there is a tendency to view it with an attitude of inertia. Hence, the use of changed tube profiles in the condenser may be considered as a long-term solution towards a rational designing of a compact heat exchanger.

A short-term, useful approach is also adopted, not from the point of view of reducing the condenser size, but on the basis of a cost-optimisation technique. A brief description of the concept is as follows :

A reduction in the tube diameters of the condenser results in augmentation in the overall heat transfer coefficient. Hence, for a given heat transfer requirement of the unit, less tube material is required and the first or procurement cost of the unit is reduced. But in consequence, the pressure drop through the tubes is increased and this requires a higher pumping power. This entails a higher pumping energy cost. Both these effects work in opposite directions, as far as the objective of minimizing the total cost of operation is concerned. Keeping in view these two contradicting requirements, the optimum tube-dimensions are determined on the basis of minimum total cost.

The same analysis is also extended to the changed tube profiles with suitable modifications.

Computer programmes in FORTRAN language are developed for the solution of all the above problems. The programmes are made very general to encompass a variety of similar problems, which can be solved with very little modification in the parent programmes. They are properly documented for easy understandability of any user, having a working familiarity with FORTRAN language. No subroutines (such as those, supplied by 'Numerical Algorithm Group,' abbreviated as 'NAG') are used in the programmes. Hence, in those cases, where accessibility to sophisticated computing systems like DEC-10 is not available, IBM personal computers with FORTRAN facility can be used to obtain the solutions with the help of the above programmes. 'Interactive Graphics Library,' popularly known by its acronym 'IGL' subroutines are used to draw graphs and the shapes of the tube profiles.

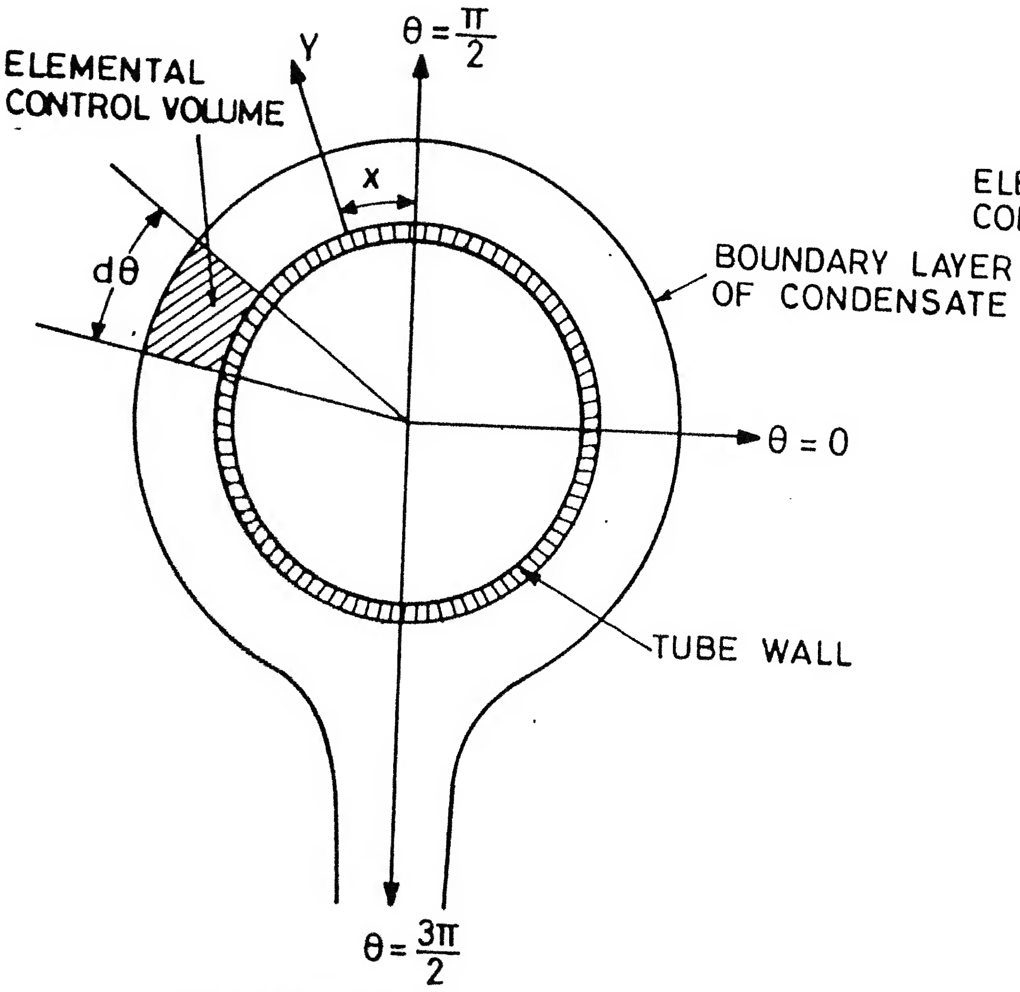
CHAPTER - II

ANALYSIS OF CONDENSATION HEAT TRANSFER AUGMENTATION BY USING PROGRESSIVELY INCREASING RADIUS OF CURVATURE PROFILE TUBES.

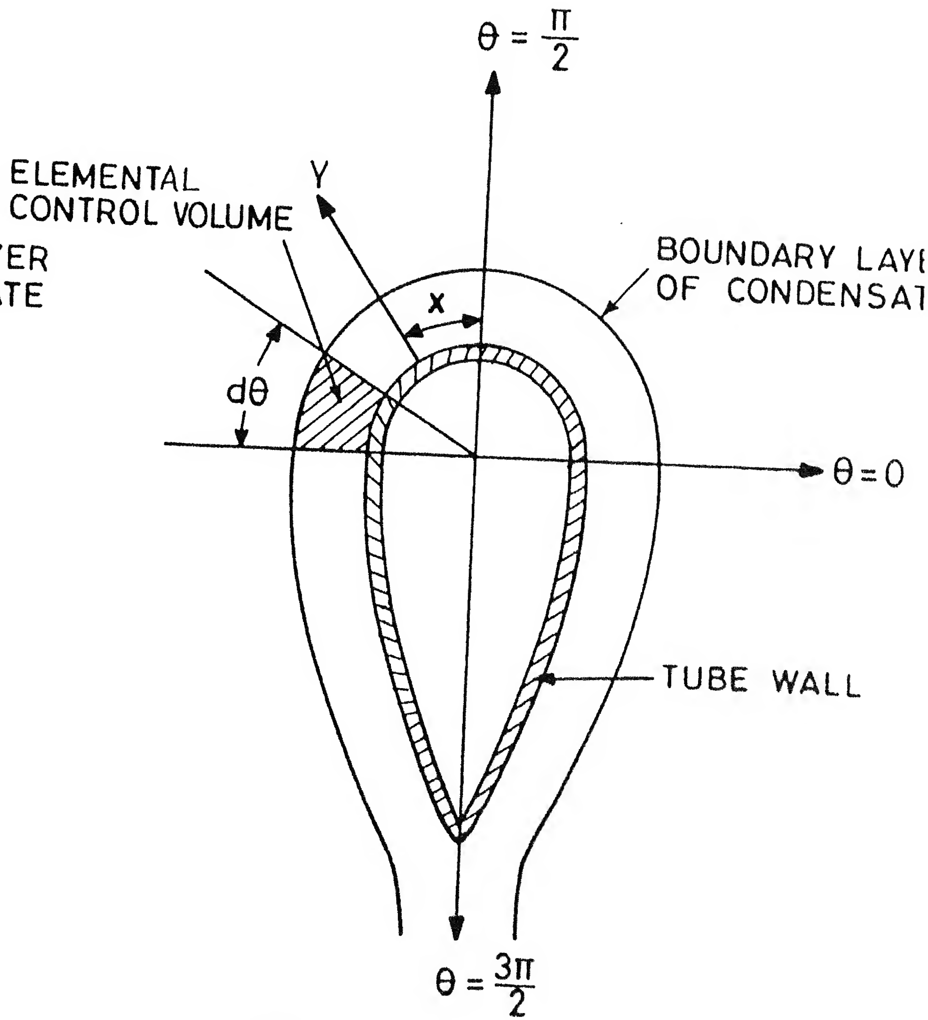
2.1 INTRODUCTION :

One of the major heat transfer resistances, associated with the process of steam, condensing over horizontal tubes of a condenser, is that of the condensate layer, formed over the tubes. Hence, heat transfer with film condensation is limited by the thickness of the condensate layer. The growth and removal of this condensate layer depends on the forces, acting on the film. For a tube with circular profile, which is the most general in applications, the velocity of the condensate is solely determined by the projection of the gravity force along the tangent to the profile and the viscous force.

If the profile of the above circular tube is moulded in such a manner that it is drawn in the direction of the gravity vector (Fig. 2.1(b)), the average value of the projection of the gravity force along the tube periphery will be greater than that of the corresponding value for circular tube. This increases the rate of run-off of the condensate and the heat transfer coefficient is, consequently, increased.



(a) CIRCULAR PROFILE



(b) P.I.R.C. PROFILE

FIG. 2 1 DIFFERENTIAL ELEMENT OF CONDENSATE IN THE BOUNDARY LAYER AND THE COORDINATE SYSTEM.

In addition to the above modification, if the moulded tube profile is made to have its radius of curvature, continuously increasing from the upper generatrix to the lower generatrix, then, since the surface-tension force is inversely proportional to the radius of curvature, a pressure gradient develops along the tube surface. Since, the pressure constantly decreases towards the lower generatrix, it acts favourably to enhance the condensate drain-out rate. The circular profile, having the same radius of curvature throughout, does not develop any pressure gradient and, thus, cannot take advantage of the effect of a favourable pressure gradient.

Hence, this discussion reveals the fact that the P.I.R.C. (progressively increasing radius of curvature) tubes (Fig. 2.1.(b)), by making use of the <u>increased gravity</u> force and a favourable <u>pressure gradient</u> (due to surface-tension), result in a faster condensate drain-out rate from over the tube surfaces, when compared with their circular counterpart. Consequently, the heat transfer coefficient registers an increase and for a given condenser heat transfer requirement, a relatively smaller heat exchanger can serve the purpose.

In this chapter, details of the solution method are presented for three different tube profiles; viz.;

(a) Circular

(b) Polar form '$r=a\theta$', being reflected on both sides of its vertical axis.

(c) Polar form '$r=ae^{\theta}$', being reflected on both sides of its vertical axis.

The solution for circular profile is already available from Nusselt's analysis. The present solution, then, is compared with the available result to vindicate the impeccability of the method, employed. Profiles (b) and (c) are chosen in line with the above discussion to take advantage of the gravity and surface-tension effects to augment the heat transfer rate in the condenser. The percentage increase in the average steam-side heat transfer coefficient, using (b) and (c) in comparison to (a) is determined. The isolated and combined effects of gravity and surface-tension on the augmentation of condensation heat transfer coefficient are presented, separately. The savings in first-cost, accruing from a smaller size condenser using P.I.R.C. profiles (b) and (c) in comparison with the presently used circular profiles are also determined.

2.2 ASSUMPTIONS, MADE IN THE ANALYSIS AND THEIR VALIDITY :

The actual process of condensation is very complex. In order to make the problem of film-condensation solvable by analytical technique, certain simplifying assumptions are made. These assumptions make the solution of the problem tractable and still preserve its realistic nature. Despite the complexities associated with film condensation, useful results are obtained by making the following assumptions.

1. The flow of the condensate film is laminar. It has been found practically [18] that the turbulent flow of the condensate is difficult to achieve on horizontal tubes, but may be easily established over the lower portions of vertical surfaces. Since, in this analysis, condensation over horizontal tubes is considered, the assumption is sufficiently reasonable.

2. The steam to be condensed is at a uniform temperature T_{sat}. There is no temperature gradient in the vapour boundary layer, formed over the liquid condensate layer. But in actual practice, a temperature difference in the vapour boundary layer is observed to manifest. However, its magnitude is of the order of $0.03^{o}C$ (for steam), which for all practical purposes can be neglected. Moreover, it

has been observed [21] that the thickness of the vapour boundary layer is of the order of 2 to 3 times the molecular diameter. For steam, the thickness is of the order of only one molecular diameter. The consideration of these microscopic dimensions in an analysis, based on the continuum approach, loses significance. Both these effects provide sufficient ground to validate the assumption. Hence, heat transfer to the liquid-vapour interface can occur only by condensation at the interface and not by conduction from the vapour.

3. The shear stress at the liquid-vapour interface is assumed to be negligible. This assumption is true because of the low steam velocities (< 170 m./sec.), commonly encountered in Vapour space condensation [21] which prevents the interfacial drag from becoming of any consequential importance.

4. The convective terms in the momentum and energy transfer in the condensate film are assumed to be negligible. This assumption is true because of the very low magnitude of velocities (of the order of 10^{-6} m./Sec.), encountered by the condensate film. Such low velocities are inconsequential from the point of view of engineering applications. Hence, the boundary layer flow of the condensate is one of very slow motion, commonly termed as the 'creeping motion.'

5. The surface temperature of the tube on the steam-side is assumed to remain constant. Since, the magnitude of heat-transfer coefficient of condensation on the steam-side is very large, the relative order of magnitude of this coefficient requires an almost negligible temperature difference for heat transfer. Moreover, actual experimental observations by Peck and Reddy and by Bromley, Brodkey and Fishman separately, corroborate the validity of this assumption.

6. Alongwith these assumptions, Nusselt's modification of the analysis for condensation over 'n' vertical tubes in a row and Chen's modification for the sub-cooling of the condensate film are also assumed to apply. This assumption is valid for those cases where Jakob No. 'Ja' is less than 2. In the present case, 'Ja' has been calculated out to be 0.0035 and thus speaks for the validity of the assumption. This assumption is made to make the analysis more realistic.

2.3 THE TUBE-WALL SURFACE TEMPERATURE:

The average steam-side heat transfer coefficient for circular profile is given by the well-known relation [5],

$$\bar{h}_s = (0.728) \times \left[1 + 0.2 \frac{C_{p_l} (T_{sat.} - T_w)}{h_{fg}} (n-1) \right] \times \left[\frac{\rho_l^2 g \ k_l^3 \ h_{fg}'}{n \ d_o \ \mu_l (T_{sat.} - T_w)} \right]^{1/4} \tag{2.1}$$

The above relation is valid when film-condensation of a vapour is taking place over 'n' rows of vertical tubes of a condenser and the effect of condensate sub-cooling is also considered.

In expression (2.1), the properties 'k_l', 'ρ_l', 'C_{p_l}' and 'μ_l' are determined at the reference temperature '$T_{ref.}$' [5], where,

$$T_{ref.} = T_w + 0.25 \ (T_{sat.} - T_w) \tag{2.2}$$

Knowledge of the tube-wall surface temperature 'T_w' is essential in order to determine the heat transfer coefficient '$\bar{h}_s$'. The following procedure is adopted to obtain 'T_w'.

Considering a set of two tubes (one each for entrance and exit of cooling water) of 7.5m. length* each, which are presently used in the condenser, taken for the case-study and using 'Appendix-1' for the operating data of the above mentioned

---

* The procedure is applicable to any general length of the condenser tube.

condenser, the heat transfer rate '$\dot{Q}$' is given by,

$$\dot{Q} = 21890 \text{ W}$$

Also, '$\dot{Q}$' is expressed as,

$$\dot{Q} = \bar{h}_s A_o (T_{sat.} - T_w) \qquad (2.3)$$

In the above expression, 'A_o' and '$T_{sat.}$' can be obtained using Appendix -1.

Expressions (2.1) to (2.3) for '$\bar{h}_s$', '$T_{ref.}$' and '$\dot{Q}$', all depend on 'T_w'. It is, therefore, necessary to first determine 'T_w'. This has been achieved as follows :

Variation of the thermophysical properties 'ρ_l', 'C_{p_l}', 'k_l' and 'μ_l' with temperature are tabulated and given in 'Appendix -2'. A 'computer programme No.1', which uses the method of least squares , has been developed to obtain polynomial relations for these thermophysical properties in terms of the condensate temperature. 'Appendix -3' lists the 'computer programme No.1' and 'Appendix-4' gives the polynomial relations. These polynomial relations have been substituted for the thermophysical properties in expression (2.1) for determining '$\bar{h}_s$'. A 'computer programme No. 2', listed in 'Appendix -5' has been developed as an iterative tool to evaluate 'T_w'.

Initially, a guess value of 'T_w' is fed into the 'computer programme No.1'. With this guess value, '$\bar{h}_s$' is determined using expression (2.1.). '$\dot{Q}$' is then calculated by using expression (2.3). This value of '$\dot{Q}$' is then matched with the actual magnitude of '$\dot{Q}$', which is <u>21890</u> W. The difference between the two is noted. A new guess-value is assigned to 'T_w' and the procedure is repeated till the difference between the actual and calculated values of '$\dot{Q}$' is negligible.

'Table 2.1' gives results for 'T_w' and some of the related properties of the condensate, corresponding to this value of 'T_w'.

2.4 GENERAL FORMULATION OF THE PROBLEM TO DETERMINE THE CONDENSATION HEAT TRANSFER COEFFICIENT:

The problem of determining the heat-transfer coefficients for the circular and polar-profile-tubes requires specifying the governing differential equations and the necessary boundary and/or initial conditions. The coordinate system used is shown in 'figure 2.1'. The effort, here, is to first derive expression of the average heat transfer coefficient for the circular tube by solving the general problem and comparing the results, so obtained, with the well-known correlation (2.1) for '$\bar{h}_s'$'. The same analysis will be extended to derive expressions for average heat transfer coefficients for the polar-profile-tubes.

The general momentum and energy conservation equations, in their differential form, are, respectively, given as following :

$$\rho_1 \left(\frac{\partial \underline{V}}{\partial \tau} + \underline{V} \cdot \underline{\nabla}\, \underline{V} \right) = \mu_1 \nabla^2 \underline{V} - \underline{\nabla} p + \underline{X} \qquad (2.4)$$

$$\rho_1 \left[\frac{\partial}{\partial \tau} (C_{p_1} T) + \underline{V} \cdot \underline{\nabla} (C_{p_1} T) \right] = - p(\underline{\nabla} \cdot \underline{V}) + \underline{\nabla} \cdot (k_1 \underline{\nabla} T) + \mu_1 \phi + \dot{Q}_g \qquad (2.5)$$

Using the assumptions of steady flow, creeping motion and Prandtl's laminar boundary layer approximations, the momentum equation reduces to ,

$$0 = \mu_1 \frac{\partial^2 u}{\partial y^2} - \frac{\partial p}{\partial x} + \underline{X} \qquad (2.6)$$

The energy transfer across the condensate-film occurs only by conduction as discussed in Section (2.2). Moreover, with the assumptions of 'steady' and 'creeping' motion for the condensate-film, expression (2.5) boils down to ,

$$0 = \underline{\nabla} \cdot (k_1 \ \underline{\nabla} \ T) \qquad (2.7)$$

Assuming constant property of the condensate and using Prandtl's laminar boundary layer approximations, the final form of energy equation is given by,

$$k_1 \ x \ \frac{\partial^2 T}{\partial y^2} = 0$$

where, k_1 is the conductivity of the condensate and is assumed constant.

$$\therefore \quad \frac{\partial^2 T}{\partial y^2} = 0 \qquad (2.8)$$

Solution of the momentum equation for obtaining the velocity distribution in the condensate boundary-layer, requires the following two boundary conditions :

$u(y) = 0, \quad y = 0$ (Condition of no slip)

$\frac{\partial}{\partial y} u(y) = 0, \; y=\delta$ (Zero interfacial shear stress)

Similarly, for the determination of temperature distribution in the condensate boundary layer, the energy equation (2.8) is used alongwith the following two boundary conditions :

$T = T_w, \quad y=0$ (Constant wall-surface temperature)

$T = T_{sat.}, \quad y=\delta$ (Constant vapour-liquid interface-temperature).

Since 'u' and 'T' are functions of 'y' only and 'p' is a function of 'x' only, the partial differentials are replaced by total differentials and the formulation of the problem is given by :

(1) The momentum equation :

$$\frac{d^2u}{dy^2} - \frac{1}{\mu_l}\frac{dp}{dx} + \frac{1}{\mu_l}\,\underline{X} = 0 \qquad (2.9)$$

Boundary conditions :

$$u = 0, \quad y = 0 \qquad (2.10)$$

$$\frac{du}{dy} = 0 \;, \quad y=\delta \qquad (2.11)$$

(2) The energy equation :

$$\frac{d^2T}{dy^2} = 0 \qquad (2.12)$$

Boundary conditions :

$$T = T_w, \ y = 0 \qquad (2.13)$$

$$T = T_{sat.}, y = \delta \qquad (2.14)$$

Expressions (2.9) to (2.14) represent the complete formulation of the problem.

2.5 SOLUTIONS FOR CONDENSATION HEAT TRANSFER COEFFICIENT FOR SPECIFIC TUBE-PROFILES :

In the absence of any specific method known for the solution of the above mentioned problem, a numerical technique has been developed for the solution of the problem and is tested for the circular profile case for which the results are known. The method is then, extended to determine heat transfer coefficients for the polar profile tubes.

2.5.1 Circular Profile :

The velocity distribution in the condensate boundary layer is obtained by solving the momentum equation. In the generalised momentum equation (2.9), the following modifications are made.

(a) Since, the surface-tension contribution is absent for circular profiles, its presence in the momentum equation is dropped.

(b) The body-force component is contributed by gravity only. Resolving the gravity vector along the tangent to the tube-profile (Fig.2.1), the magnitude of the body-force '$\underline{X}$' is given by,

$$\underline{X} = \rho_1 g \, \mathrm{Sin}\, \beta$$

$$= \rho_1 \, g \, \mathrm{Sin}(\theta - 90^\circ)$$

$$= \rho_1 \, g \, (-\mathrm{Cos}\theta)$$

$$\therefore \quad \underline{X} = \rho_1 \, g \, (-\mathrm{Cos}\theta) \tag{2.15}$$

With the above modifications, the expression (2.9) becomes :

$$\frac{d^2u}{dy^2} + \frac{\rho_1 g}{\mu_1} (-\mathrm{Cos}\ \theta) = 0 \tag{2.16}$$

If,

$$\frac{\rho_1 g}{\mu_1} (-\mathrm{Cos}\ \theta) = F(\theta) \tag{2.17}$$

Then the momentum equation becomes :

$$\frac{d^2u}{dy^2} + F(\theta) = 0 \tag{2.18}$$

Integrating the equation (2.16) twice and using the boundary conditions, given by expressions (2.10) and (2.11), the velocity distribution is obtained as :

$$u = \frac{\rho_1 g}{\mu_1} (-\mathrm{Cos}\ \theta) \left[\delta\, y - \frac{y^2}{2} \right] \tag{2.19}$$

The mean velocity 'u_m' at any location along the periphery of the tube is then given by,

$$u_m = \frac{1}{\delta} \int_0^\delta u \, dy \tag{2.20}$$

where, 'δ' is the condensate-boundary layer thickness. Simplification of expression (2.20) yields,

$$u_m = \frac{\rho_l g \, \delta^2}{3\mu_l} \, (-\mathrm{Cos}\ \theta) \qquad (2.21)$$

The temperature distribution is, then, obtained by integrating equation (2.12) twice and using the boundary conditions, given by expressions (2.13) and (2.14). This gives :

$$T = \frac{T_{sat.} - T_w}{\delta} \, .y + T_w \qquad (2.22)$$

Consider now a control volume of unit thickness, indicated by the hatched portion of Fig. (2.1.a), which is contained by the infinitesimal angle '$d\theta$'.

An energy balance of this control volume gives :

$$dm \times h_{fg} = \frac{k_l}{\delta} \, (T_{sat.} - T_w) \times (dx \times 1) \qquad (2.23)$$

where,

dm = the net mass efflux from the control volume.

$$= d \left[\rho_l \, u_m \, (\delta x 1) \right] \qquad (2.24)$$

Substitution of the expression (2.21) for 'u_m' in (2.24) yields,

$$dm = d \left[\frac{\rho_l^2 \, g \, \delta^3}{3\mu_l} \, (-\mathrm{Cos}\ \theta) \right] \qquad (2.25)$$

h_{fg} = Latent heat of evaporation

dx = Elemental peripheral length

$= R\, d\theta$, 'R' being the outer radius of the circular profile tube. (2.26)

But,

$$R = \frac{d_o}{2} \tag{2.27}$$

where, 'd_o' is the outer diameter of the circular tube.

Using (2.27), the expression (2.26) for 'dx' reduces to :

$$dx = \left(\frac{d_o}{2}\right) d\theta \tag{2.28}$$

It is to be noted that in expression (2.25), the boundary layer thickness 'δ' is a function of 'θ'.

Substituting for 'dm' and 'dx' in (2.23), we get,

$$d\left[\frac{\rho_1^2\, g\, \delta^3}{3\mu_1}(-\mathrm{Cos}\ \theta)\right] x\, h_{fg} = \frac{k_1}{\delta}(T_{sat.} - T_w)\left(\frac{d_o}{2}\right) d\theta \tag{2.29}$$

Simplification of (2.29) gives,

$$d\left[\frac{\rho_1^2\, g\, h_{fg}}{\mu_1\, k_1\, d_o (T_{sat.} - T_w)}\left(-\frac{2}{3}\,\mathrm{Cos}\ \theta\right)(\delta^3)\right] = \frac{1}{\delta}\, d\theta \tag{2.30}$$

In the above expression, the condensate properties are evaluated at reference temperature '$T_{ref.}$' by using Appendix- 4

If,

$$\frac{\rho_l^2\ g\ h_{fg}}{\mu_l\ k_l\ d_o(T_{sat.}-T_w)} = N = \text{Constant} \qquad (2.31)$$

and

$$(-\tfrac{2}{3}\ \text{Cos}\ \theta) = F_1(\theta) \qquad (2.32)$$

the equation (2.30), may be written as,

$$(N)\ x\ d\left[F_1(\theta).\ \delta^3\right] = \frac{1}{\delta}\ .\ d\theta$$

or

$$(N)x\ \delta\ x\ d\left[F_1(\theta)\ \delta^3\right] = d\theta \qquad (2.33)$$

Multiplying both side of expression (2.33) by $\left\{F_1(\theta)\right\}^{1/3}$ and simplifying, the following result is obtained :

$$(N)\ x\ d\left[\frac{3}{4}\ x\ \delta^4 x\ \ F_1(\theta)^{4/3}\right] = \left\{F_1(\theta)\right\}^{1/3} d\theta \qquad (2.34)$$

Integration of equation (2.34) and simplification gives :

$$\delta = \left(\frac{4}{3}\right)^{1/4} x\ \frac{\left[\int_{\theta=\pi/2}^{\theta}\left\{F_1(\theta)\right\}^{1/3} d\theta\right]^{1/4}}{\left\{F_1(\theta)\right\}^{1/3}} x\ \frac{1}{(N)^{1/4}} \qquad (2.35)$$

The steam-side heat transfer coefficient 'h_s'' at any location 'θ' is given by [19],

$$h_s = \frac{k_1}{\delta} \qquad (2.36)$$

Substitution of expression (2.35) for 'δ' in expression (2.36) gives :

$$h_s=(k_1)x(\tfrac{3}{4})^{1/4} \times \frac{\left\{F_1(\theta)\right\}^{1/3}}{\left[\int_{\theta=\pi/2}^{\theta}\left\{F_1(\theta)\right\}^{1/3} d\theta\right]^{1/4}} \times (N)^{1/4}$$

or

$$h_s=(\tfrac{3}{4})^{1/4} \times \frac{\left\{F_1(\theta)\right\}^{1/3}}{\left[\int_{\theta=\pi/2}^{\theta}\left\{F_1(\theta)\right\}^{1/3} d\theta\right]^{1/4}} \times (k_1^4\ N)^{1/4}$$

(2.37)

Using expression (2.31) for 'N' in (2.37), we get,

$$h_s=(\tfrac{3}{4})^{1/4} \times \frac{\left\{F_1(\theta)\right\}^{1/3}}{\left[\int_{\theta=\pi/2}^{\theta}\left\{F_1(\theta)\right\}^{1/3} d\theta\right]^{1/4}} \times \left[\frac{\rho_1^2\ g\ h_{fg}\ k_1^3}{\mu_1\ d_o(T_{sat.} - T_w)}\right]^{1/4}$$

(2.38)

Relation (2.38) is derived for steam, condensing over a single circular, horizontal tube. But in an actual condenser, condensation occurs over a number of tubes and also the condensate is subcooled. To take account of these effects,

Nusselt's correction for condensation over 'n' horizontal tubes in a vertical row and Chen's correction factor for condensate subcooling [5] are applied to expression (2.38). The application of these correction factors makes the analysis more realistic. With these modifications, expression (2.38) for 'h_s'' reduces to ,

$$h_s = \left(\frac{3}{4}\right)^{1/4} \times \underbrace{\frac{\left\{F_1(\theta)\right\}^{1/3}}{\left[\int_{\theta=\pi/2}^{\theta} \left\{F_1(\theta)\right\}^{1/3} d\theta\right]^{1/4}}}_{\mathrm{I}} \times \underbrace{\left[\frac{\rho_l^2\, g\, h_{fg}'\, k_l^3}{n\, d_o\, \mu_l (T_{sat.} - T_w)}\right]^{1/4}}_{\mathrm{II}}$$

$$\times \underbrace{\left[1 + 0.2\, \frac{C_{p_l}\, (T_{sat.} - T_w)}{h_{fg}}\, (n-1)\right]}_{\mathrm{III}} \qquad (2.39)$$

where,

n = Average number of tubes in one vertical row of the condenser.

h_{fg}' = Corrected latent heat of evaporation, considering condensate subcooling = $h_{fg} + \frac{3}{8}\, C_{p_l} (T_{sat.} - T_w)$

It is to be noted that in the expression (2.39) for 'h_s'' , part-I is a function of 'θ', whereas part-II and part-III are constants. Evaluation of the integral in expression (2.39) analytically, is a formidable task. In order to overcome this difficulty, the following method is employed :

The semi-perimeter of the profile is divided into 100 equal parts. At each discrete location of 'θ', the local heat transfer coefficient 'h_s' is evaluated with the help of expression (2.39), the integral being obtained by using 'Simpson's numerical rule of integration'. Since at '$\theta = \pi/2$' and '$\theta = 3\pi/2$', the expression (2.39) assumes the indeterminate (zero/zero) form, these values are suitably truncated to circumvent this difficulty. for example, '$\theta = \frac{\pi}{2}$' is truncated to '$\theta = \frac{\pi}{2} + 0.01\pi = 1.6022$' and '$\theta = \frac{3\pi}{2}$' is truncated to '$\theta = \frac{3\pi}{2} - 0.01\pi = 4.6809$'. A 'computer programme No. 3' is given in 'Appendix -6', which has been used to calculate 'h_s' at discrete locations 'θ'. Some representative values of 'h_s' vs. 'θ', obtained from this computation are presented in 'Table 2.2'. With these values of 'h_s' and 'θ' at discrete locations, a polynomial relation between the two variables is obtained by using 'the method of least squares'. A 'computer programme' No.4, listed in 'Appendix-7', has been used to obtain the polynomial relation between 'h_s' and 'θ'. The constant terms, viz. part-II and part-III in expression (2.39) are kept unaltered and the part-I of this expression (2.39) only generates the polynomial relation, which is computed, using the above mentioned procedure.

This polynomial relation between 'h_s' and 'θ' comes out to be ,

$$h_s = \left[1.3324 - 0.4932\,\theta + 0.1946\,\theta^2 - 0.0293\,\theta^3\right] \times \left[K\right] \tag{2.40}$$

$$K = \left[\left\{ 1+0.2\,\frac{C_{p_1}(T_{sat.}-T_w)}{h_{fg}}\,(n-1) \right\} \times \left\{ \frac{\rho_1^2\, g\, h_{fg}\, k_1^3}{n\, d_o\, \mu_1 (T_{sat.}-T_w)} \right\}^{1/4} \right] \tag{2.41}$$

Hence, 'K' is the product of the two constant terms, viz. part-II and part-III of expression (2.39).

The average steam-side heat transfer coefficient '$\bar{h}_s$' is then given by,

$$\bar{h}_s = \frac{\int_{\theta_1}^{\theta_2} h_s\, d\theta}{(\theta_2-\theta_1)} \tag{2.42}$$

Substituting expression (2.40) for 'h_s' in (2.42) and evaluating the integral between the truncated limits 'θ_1=1.6022' and 'θ_2= 4.6809' , we get the desired expression for the average steam-side heat transfer coefficient,

$$\bar{h}_s = (0.7303) \times [K] \tag{2.43}$$

where 'K' is given by expression (2.41).

The standard result for '$\bar{h}_s$' , for circular profile, given by expression (2.1) may be represented as :

$$\bar{h}_s = (0.728) \times [K] \tag{2.44}$$

Comparison of the derived expression (2.43) for '$\bar{h}_s$' with the standard result (2.44), it may be easily seen that the percentage error between the calculated and known expressions for 'h_s', in case of circular profile is given by,

$$\text{\% error} = \frac{\text{'}\bar{h}_s\text{' from expression (2.43)} - \text{'}\bar{h}_s\text{' from expression (2.44)}}{\text{'}\bar{h}_s\text{' from expression (2.44)}} \times 100$$

$$= \frac{(0.7303)\times[K] - (0.728)\times[K]}{(0.728)\times K} \times 100$$

$$= \frac{0.7303 - 0.728}{0.728} \times 100$$

$$= 0.288\ \%$$

Since, the error is negligible, the calculated result is in close agreement with the known result. This proves the efficacy of the method, employed for the solution of the problem.

2.5.2 Profile Represented by polar curve, '$r = a\theta$', Reflected symmetrically on both sides of its vertical axis :

Fig. (2.2.a) shows the actual shape of the profile '$r = a\theta$.' The corresponding circular profile, which has been moulded to generate the above polar profile is also shown in Fig. (2.2.b). A 'Computer programme No.5 ', listed in 'Appendix-8,' has been used to draw these profiles. The same co-ordinate system as that of Section (2.5.1) is used.

Some geometrical relations, concerning the profile are evaluated first, which aid in the formulation and solution of the problem.

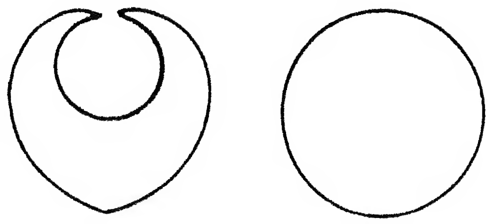


FIG. 2.2 ACTUAL SHAPES OF THE POLAR PROFILE '$r = a\theta$' AND THE CORRESPONDING CIRCULAR PROFILE.

SEMI PERIMETER

A differential length 'dx' along the periphery of the profile is given by,

$$dx = \sqrt{r^2 + \left(\frac{dr}{d\theta}\right)^2} \qquad (2.45)$$

Here,

$$r = a\theta \qquad (2.46)$$

$$\therefore \quad \frac{dr}{d\theta} = a \qquad (2.47)$$

Substitution of (2.46) and (2.47) in (2.45) gives,

$$dx = a\sqrt{1 + \theta^2} \quad d\theta \qquad (2.48)$$

The semi perimeter 'S_p' is represented as :

$$S_p = \int_{\theta=\pi/2}^{3\pi/2} a\sqrt{1 + \theta^2} \quad d\theta \qquad (2.49)$$

or

$$S_p^* = \frac{1}{2} a \left[\left\{ \theta\sqrt{1+\theta^2} \right\} + \left\{ \ln \left(\theta + \sqrt{1+\theta^2} \right) \right\} \right] \Bigg|_{\theta=\pi/2}^{\theta=3\pi/2}$$

$$(2.50)$$

On simplification, expression (2.50) gives :

$$S_p = 10.3985 \ a \qquad (2.51)$$

---

* Results of standard integration table have been used :

$$\int \sqrt{x^2 + a^2} \quad dx = \frac{1}{2} x \sqrt{a^2 + x^2} + \frac{1}{2} a^2 \sinh^{-1}(x/a)$$

$$\sinh^{-1}(x/a) = \ln \left\{ x + \sqrt{x^2 + a^2} \right\}$$

Since, this semiperimeter is kept equal to the semi perimeter of the corresponding circular profile, it leads to the following expression :

$$\pi R = S_p = 10.3985\ a$$

$$\therefore \quad a = \frac{\pi}{10.3985} R = 0.3021\ R \qquad (2.52)$$

In the above expression, the factor '0.3021' is named as 'radius multiplying factor' or 'RMF'. Multiplying 'RMF' by the radius of the circular profile, results in the determination of the characteristic constant 'a' of the polar profile, '$r = a\theta$'.

AREA OF CROSS-SECTION

The cross-sectional area of the circular profile, which is moulded into the polar profile under consideration, is given by ,

$$(A_{cn})_c = \frac{\pi}{4}\ d_i^2 \qquad (2.53)$$

where,

subscript 'c' stands for 'circular.'

d_i = Inside diameter of the circular tube

A_{cn} = Area of cross-section

From the relation (2.52), the constant 'a' can be expressed as :

$$a = (0.3021) \times \left(\frac{d_i}{2}\right) \qquad (2.54)$$

The area of cross-section of the polar profile is given by,

$$(A_{cn})_p = \int_{\theta=\pi/2}^{3\pi/2} r^2 \, d\theta$$

$$= a^2 \int_{\theta=\pi/2}^{3\pi/2} \theta^2 \, d\theta$$

$$= a^2 \left[\frac{\theta^3}{3}\right]_{\theta=\pi/2}^{3\pi/2}$$

$$= (0.3021)^2 \times \left(\frac{26\pi^3}{24}\right) \times \left(\frac{d_i}{2}\right)^2$$

$$\therefore \quad (A_{cn})_p = (0.3021)^2 \times \left(\frac{26\pi^3}{24}\right) \times \left(\frac{d_i}{2}\right)^2 \qquad (2.55)$$

In expression (2.55), the subscript 'p' stands for 'polar.'

An area factor 'AF' is now defined, which, when multiplied by the cross-sectional area of the circular profile, yields the cross-sectional area for the corresponding polar profile. In the present case, 'AF' is calculated as :

$$AF = \frac{(A_{cn})_p}{(A_{cn})_c} = \frac{\text{expression (2.55)}}{\text{expression (2.53)}} = \underline{0.9758} \qquad (2.56)$$

HYDRAULIC MEAN DIAMETER

The perimeters of the circular and polar profiles are kept equal to provide the same heat transfer surface area for both the profiles. Hence, the perimeter 'PER' of the polar profile, '$r=a\theta$,' is given by,

$$PER = \pi\ d_i \tag{2.57}$$

The hydraulic mean diameter 'HMD' is then obtained using :

$$HMD = \frac{4 \times AF \times (A_{cn})_c}{PER} \tag{2.58}$$

RADIUS OF CURVATURE

The radius of curvature 'R_c' of the profile at any location, is evaluated as follows :

$$R_c = \frac{\left\{ r^2 + \left(\frac{dr}{d\theta}\right)^2 \right\}^{3/2}}{r^2 + 2\left(\frac{dr}{d\theta}\right)^2 - r\left(\frac{d^2r}{d\theta^2}\right)} \tag{2.59}$$

Here,

$$r = a\theta, \quad \frac{dr}{d\theta} = a\ , \quad \frac{d^2r}{d\theta^2} = 0$$

$$\therefore \qquad R_c = \frac{(a^2\theta^2 + a^2)^{3/2}}{a^2\theta^2 + 2a^2}$$

or,

$$R_c = a\left[\frac{(1+\theta^2)^{3/2}}{(2+\theta^2)}\right] \tag{2.60}$$

SURFACE-TENSION PRESSURE

The surface-tension pressure at any location is inversely proportional to the radius of curvature at that location and is given by,

$$p = \frac{\sigma_1}{R_c} = \frac{\sigma_1}{a} \left[\frac{(2 + \theta^2)}{(1+\theta^2)^{3/2}} \right] \tag{2.61}$$

where,

σ_1 = Surface-tension force per unit length.

The pressure gradient along the profile is, then, determined as given below :

$$\frac{dp}{dx} = \left(\frac{dp}{d\theta} \right) / \left(\frac{dx}{d\theta} \right) \tag{2.62}$$

Using (2.48),

$$\frac{dx}{d\theta} = a \sqrt{1+ \theta^2} \tag{2.63}$$

Differentiating expression (2.61) with respect to 'θ' and with the help of (2.63), the expression (2.62) gives :

$$\frac{dp}{dx} = - \frac{\sigma_1}{a^2} \left[\frac{(\theta^3+4\theta)}{(1+\theta^2)^3} \right] \tag{2.64}$$

The negative sign of this pressure gradient is in conformity with the expectation of a favourable pressure gradient.

Having completed the evaluation of the above relations, we now proceed to the formulation and solution of the problem.

The governing equations of the general formulation, given in Section (2.4) are now modified as following :

(a) The gravity force is again the same as that for the case of circular profile, i.e. '$\underline{X} = \rho_1 \, g(-\mathrm{Cos}\ \theta)$';

(b) The surface-tension pressure gradient is now given by the expression (2.64), i.e.

$$\frac{dp}{dx} = - \frac{\sigma_1}{a^2} \left[\frac{(\theta^3 + 4\theta)}{(1+\theta^2)^3} \right]$$

Using (a) and (b) in the momentum equation (2.9), we get,

Momentum equation :

$$\frac{d^2u}{dy^2} + \frac{\rho_1 g}{\mu_1} (-\text{Cos }\theta) + \frac{\sigma_1}{\mu_1 a^2} \left\{ \frac{(\theta^3+4\theta)}{(1+\theta^2)^3} \right\} = 0 \tag{2.65}$$

Boundary conditions :

$$u = 0 , \quad y = 0$$

$$\frac{du}{dy} = 0 , \quad y = \delta$$

Energy equation is the same as that for the case of circular profile. It can be easily seen that in the above formulation of the problem for polar profile, '$r=a\theta$', everything is the same as the formulation for the circular profile (Section 2.5.1) except the momentum equation (2.65). Hence, the momentum equation is suitably modified to fit into the pattern of the formulation for circular profile in the following manner :

Rearrangement of terms in expression (2.65) gives :

$$\frac{d^2u}{dy^2} + \left[\frac{\rho_1 g}{\mu_1} (\mathrm{Cos}\ \theta) + \frac{\sigma_1}{\mu_1 a^2} \left\{ \frac{(\theta^3 + 4\theta)}{(1+\theta^2)^3} \right\} \right] = 0 \qquad (2.66)$$

consider,

$$F_g(\theta) = \frac{\rho_1\ g}{\mu_1} (-\mathrm{Cos}\ \theta) \qquad (2.67)$$

$$F_s(\theta) = \frac{\sigma_1}{\mu_1\ a^2} \left\{ \frac{(\theta^3+4\theta)}{(1+\theta^2)^3} \right\} \qquad (2.68)$$

$$F\ (\theta) = F_g(\theta) + F_s(\theta) \qquad (2.69)$$

'$F_g(\theta)$' stands for the contribution of gravity force and '$F_s(\theta)$' for the contribution of surface-tension force to the condensate drain-off rate.

If '$F(\theta)$' of expression (2.69) now replaces '$F(\theta)$' of expression (2.18) in section (2.5.1), both formulations for the circular and the polar profile cases will be exactly alike. Therefore, the same solution method as adopted earlier for the circular profile is applied to solve the present problem.

Some representative values of 'h_s' vs. 'θ', obtained from the solution of the above problem are given in 'Table 2.3.' The effects of both gravity and surface-tension have been considered. The local heat transfer coefficient 'h_s' as a function of 'θ' is obtained as :

$$h_s = \left[3.4363 - 2.2905\theta + 0.7086\ \theta^2 - 0.0779\ \theta^3\right] \times \left[K\right] \qquad (2.70)$$

where,

'K' is given by expression (2.41)

The average steam-side heat transfer coefficient '$\bar{h}_s$' is calculated to be :

$$\bar{h}_s = (0.7991) \times [K] \qquad (2.71)$$

Percentage increase in '$\bar{h}_s$' in comparison with the corresponding circular profile is given by,

$$\% \text{ increase} = \frac{\text{'}\bar{h}_s\text{' from expression (2.71)} - \text{'}\bar{h}_s\text{' from expression (2.44)}}{\text{'}\bar{h}_s\text{' from expression (2.44)}} \times 100$$

$$= \underline{9.42\ \%} \qquad (2.72)$$

To study the isolated contribution of gravity towards augmentation in '$\bar{h}_s$' for the polar profile over '$\bar{h}_s$' for the corresponding circular profile, the following procedure is adopted.

In the expression (2.69) for '$F(\theta)$', the surface-tension contribution '$F_s(\theta)$' is set equal to zero. Then, adopting the same solution procedure, as is used for the circular profile, the following results are obtained. Values of 'h_s' vs. 'θ' are obtained at various discrete locations along the periphery of the profile. Some representative values of 'h_s' vs. 'θ' are presented in 'Table 2.4'.

'h_s' is given by,

$$h_s = \left[1.5879 - 0.5665\,\theta + 0.1925\,\theta^2 - 0.0280\,\theta^3\right] \times [K]$$

(2.73)

'$\bar{h}_s$' is evaluated to be :

$$\bar{h}_s = (0.7830) \times [K] \qquad (2.74)$$

% increase in '$\bar{h}_s$' in comparison with the circular profile is calculated as :

$$\%\ \text{increase} = \frac{0.7830 - 0.728}{0.728} \times 100 = \underline{7.22}\ \% \qquad (2.75)$$

% increase in '$\bar{h}_s$' due to the lone contribution of surface tension = (2.72) - (2.75) = 9.42 % - 7.22 % = <u>2.20</u> %

(2.76)

From expression (2.72), it is observed that using the polar profile, '$r = a\theta$', causes an increase in '$\bar{h}_s$' by <u>9.42</u> % over the corresponding circular profile. A heat transfer coefficient factor 'HTCF' is now defined as :

$$\text{HTCF} = \frac{\text{'}\bar{h}_s\text{' for the polar profile from expression (2.71)}}{\text{'}\bar{h}_s\text{' for the circular profile from expression (2.44)}}$$

$$= \frac{(0.7991) \times [K]}{(0.728) \times [K]} = \underline{1.0942} \qquad (2.77)$$

From the definition of 'HTCF', it follows that this factor, when multiplied by '$\bar{h}_s$' for circular profile, will give '$\bar{h}_s$' for the corresponding polar profile.

Therefore, '$\bar{h}_s$' for the profile '$r=a\theta$' can be represented as :

$$\bar{h}_s = (\text{HTCF}) \times (0.728) \times [K] \tag{2.78}$$

where, 'K' is given by expression (2.41)

For the polar profile, under consideration, variation of 'h_s' with 'θ' is plotted in Fig. (2.3). Values of these variables of 'Table 2.3' and 'Table 2.4' are used, respectively, to graphically represent the contribution of 'gravity + surface-tension' and the contribution of 'gravity only' to the heat transfer coefficient. Variation of 'h_s' with 'ω' for the corresponding circular profile, using 'Table 2.1', is also drawn in the same figure. A 'computer programme No.6' is given in 'Appendix-9,' which has been used to draw the above figure. A glance at Fig. (2.3) gives a feel of the relative contribution of 'gravity' and 'surface-tension' to the heat transfer augmentation in the polar profile tubes in comparison with the corresponding circular tubes.

2.5.3 Profile, Represented by polar curve '$r=ae^{\theta}$', Reflected on both sides of its vertical axis :

Fig. (2.4.a) shows the actual shape of the profile. The corresponding circular profile is also shown in Fig. (2.4.b). The method for drawing these figures is the same as that for polar profile '$r=a\theta$' (Section 2.5.2). Since, the procedure of solution is exactly the same as discussed in Section (2.5.2), only the important results, associated with the use of this profile, are presented.

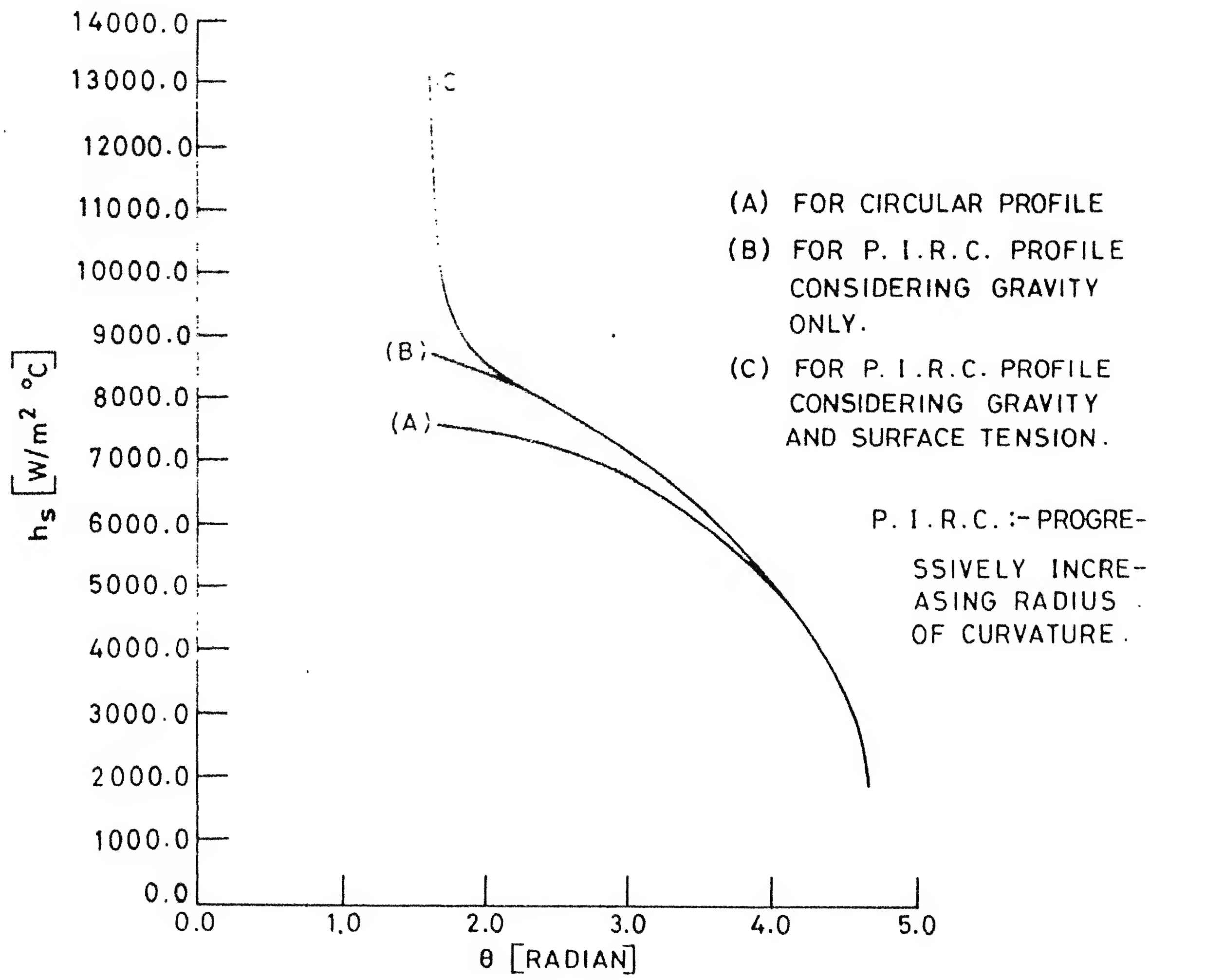
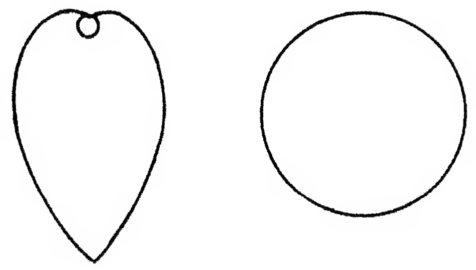


FIG. 2.3 VARIATION OF 'h_s' WITH 'θ' FOR THE POLAR PROFILE '$r = a\theta$' AND FOR THE CORRESPONDING CIRCULAR PROFILE.



(a) PROFILE '$r = ae^{\theta}$' (b) CIRCULAR PROFILE

FIG. 2.4 ACTUAL SHAPES OF THE POLAR PROFILE '$r = ae^{\theta}$' AND THE CORRESPONDING CIRCULAR PROFILE.

$$dx = \sqrt{2}\, a e^{\theta}\, d\theta \qquad (2.79)$$

$$S_p = 150.6241\ a \qquad (2.80)$$

$$RMF = 0.0209 \qquad (2.81)$$

$$AF = 0.8558 \qquad (2.82)$$

$$R_c = \sqrt{2}\, a e^{\theta} \qquad (2.83)$$

$$\frac{dp}{dx} = \frac{\sigma_1}{2a^2}\, e^{-2\theta} \qquad (2.84)$$

$$F_g(\theta) = \frac{\rho_1 g}{\mu_1}\,(-\mathrm{Cos}\ \theta) \qquad (2.85)$$

$$F_s(\theta) = \frac{\sigma_1}{2\mu_1 a^2}\, e^{-2\theta} \qquad (2.86)$$

'Table 2.5' gives some representative values of 'h_s' vs. 'θ' considering both gravity and surface-tension. For this case, 'h_s' is given by,

$$h_s = \left[15.2555 - 12.37[illegible]\theta + 3.5731\,\theta^2 - 0.[illegible]\,\theta^3\right] \times [K] \qquad (2.87)$$

'$\bar{h}_s$' is then calculated as :

$$\bar{h}_s = (1.1475) \times [K] \qquad (2.88)$$

% increase in '$\bar{h}_s$' in comparison with the corresponding circular profile

$$= 57.13\ \% \qquad (2.89)$$

Heat transfer coefficient factor = HTCF = 1.5713 (2.90)

'Table 2.6' presents some representative values of 'h_s' vs. 'θ', considering the effect of gravity only. For this case,

$$h_s = \left[2.1628 - 0.5822\,\theta + 0.1310\,\theta^2 - 0.0197\,\theta^3\right] \times [K] \quad (2.91)$$

$$\bar{h}_s = (0.9715) \times [K] \quad (2.92)$$

% increase in '$\bar{h}_s$' over the corresponding circular profile = <u>33.03</u> %. (2.93)

The surface-tension contribution towards the increase in

$$'\bar{h}_s' = 57.13\,\% - 33.03\,\% = \underline{24.10}\,\% \quad (2.94)$$

Fig. (2.5) shows the graphical representation of 'h_s' vs. 'θ' for all the above cases. The same method, as adopted for polar profile, '$r=a\theta$' in Section (2.5.2) is employed to draw this figure.

2.5.4 <u>Development of some generalised expressions</u> :

From the discussions in the Sections (2.5.1) to (2.5.3), the following generalised expressions are developed, which can be applied to any shape of the tube-profile.

(a) The local heat transfer coefficient h_s :

$$h_s = (\mathrm{HTCF}) \times (0.728) \times \left[1 + 0.2\,\frac{C_{p_l}(T_{sat.} - T_w)}{h_{fg}}(n-1)\right] \times \left[\frac{\rho_l^2\, g\, h_{fg}\, k_l^2}{n\, d_o\, \mu_l (T_{sat.} - T_w)}\right]^{1/4} \quad (2.95)$$

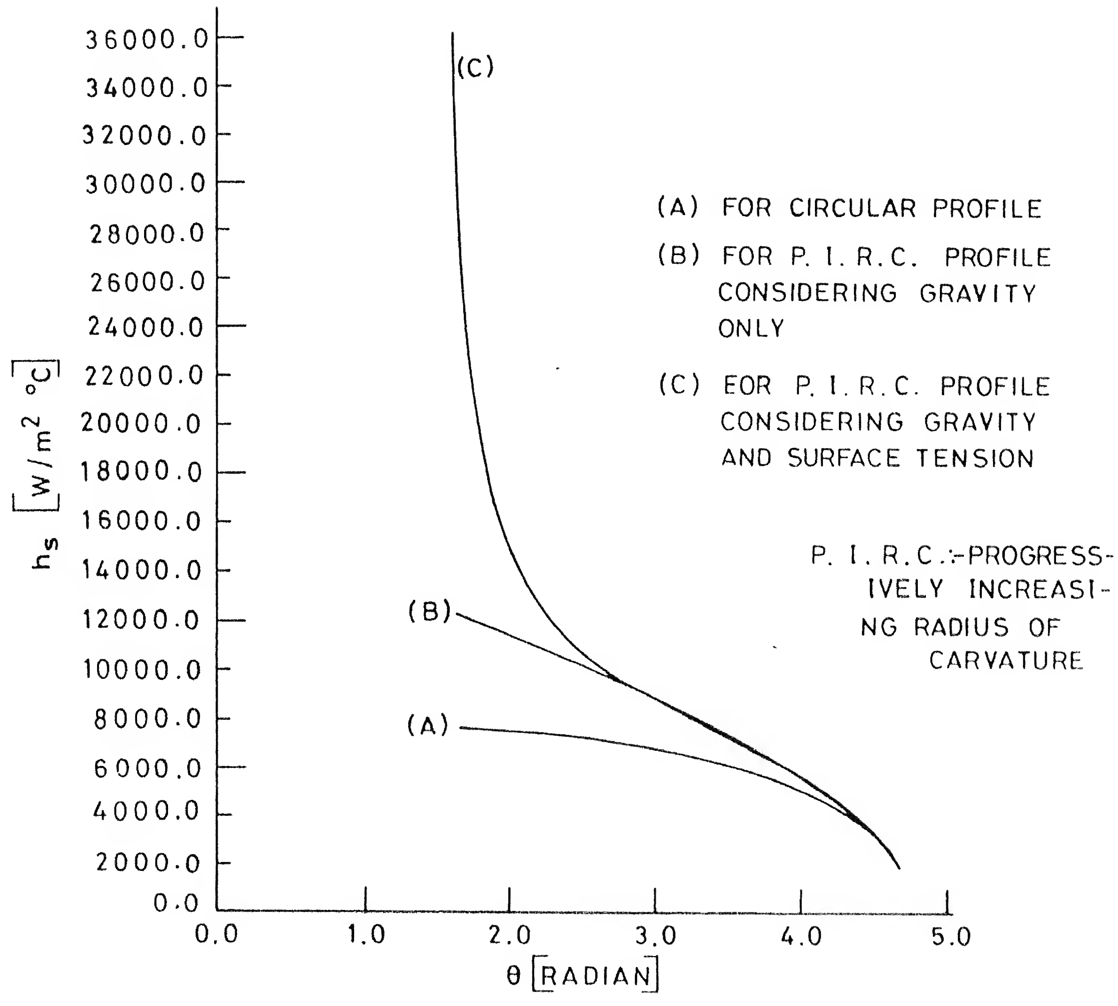


FIG. 2.5 VARIATION OF 'h_s' WITH 'θ' FOR THE POLAR PROFILE '$r=ae^{\theta}$' AND FOR THE CORRESPONDING CIRCULAR PROFILE.

In expression (2.95), 'HTCF' for different profiles are given as following

Profile	HTCF
Circular	1.0
$r=a\theta$	1.0942
$r=ae^{\theta}$	1.5713

(2.96

(b) The cross-sectional area 'A_{cn}' :

$$A_{cn} = (AF) \times (A_{cn})_c \qquad (2.97$$

In expression (2.97), '$(A_{cn})_c$' represents the cross-sectional area of the circular profile and 'AF' for different profiles are given as following :

Profile	AF
Circular	1.0
'$r=a\theta$'	0.9758
'$r=ae^{\theta}$'	0.8558

(2.98)

The above generalised relations (2.95 to 2.98) will be used in the ensuing sections for the cost optimization analysis of the condensing unit of a specific case.

Table 2.2 Values of 'h_s' at different locations 'θ' along the periphery of the condenser-tube.

(Circular profile)

Sl.No.	θ [Radian]	h_s [$W/m^2\ ^oC$]
1.	1.6022122	7623.808
2.	1.8849553	7500.937
3.	2.1991143	7386.562
4.	2.5132734	7197.425
5.	2.8274325	6928.484
6.	3.1415916	6574.778
7.	3.4557507	6125.256
8.	3.7699098	5559.154
9.	4.0840688	4829.826
10.	4.3982276	3797.853
11.	4.6809700	1745.718

Table 2.3 Values of 'h_s' at different locations 'θ' along the periphery of the condenser-tube.

(Profile, '$r=a\theta$', with gravity and surface-tension)

Sl.No.	θ [Radian]	h_s [$W/m^2\ {}^oC$]
1.	1.6022122	13098.766
2.	1.8849553	8802.841
3.	2.1991143	8234.439
4.	2.5132734	7832.231
5.	2.8274325	7407.571
6.	3.1415916	6923.467
7.	3.4557507	6363.614
8.	3.7699098	5706.693
9.	4.0840688	4908.249
10.	4.3982276	3825.664
11.	4.6809700	1856.422

Table 2.4 Values of 'h_s' at different locations 'θ' along the periphery of the condenser-tube.

(profile, '$r=a\theta$', with gravity only)

Sl.No.	θ [Radian]	h_s [W/m^2 ^oC]
1.	1.6022122	8784.459
2.	1.8849553	8490.389
3.	2.1991143	8204.885
4.	2.5132734	7854.378
5.	2.8274325	7437.283
6.	3.1415916	6949.970
7.	3.4557507	6383.691
8.	3.7699098	5719.313
9.	4.0840688	4911.908
10.	4.3982276	3824.526
11.	4.6809700	1746.612

Table 2.5 Values of 'h_s' at different locations 'θ' along the periphery of the condenser-tube.
(profile, '$r=ae^{\theta}$', with gravity and surface-tension)

Sl.No.	θ [Radian]	h_s [W/m^2 ^oC]
1.	1.6022122	36364.306
2.	1.8849553	17131.704
3.	2.1991143	12734.500
4.	2.5132734	10636.567
5.	2.8274325	9334.939
6.	3.1415916	8304.248
7.	3.4557507	7335.934
8.	3.7699098	6342.675
9.	4.0840688	5264.478
10.	4.3982276	3974.910
11.	4.6809700	1860.889

Table 2.6 Values of 'h_s' at different locations 'θ' along the periphery of the condenser-tube.
(profile, '$r=ae^{\theta}$', with gravity only)

Sl.No.	θ [Radian]	h_s [$W/m^2\ ^oC$]
1.	1.6022122	12364.641
2.	1.8849553	11674.965
3.	2.1991143	10965.827
4.	2.5132734	10178.106
5.	2.8274325	9324.292
6.	3.1415916	8414.611
7.	3.4557507	7453.368
8.	3.7699098	6433.805
9.	4.0840688	5323.377
10.	4.3982276	3999.463
11.	4.6809700	1779.375

CHAPTER-III

COST-BASED OPTIMISATION OF THE CONDENSER TUBE-DIMENSIONS FOR CIRCULAR AND POLAR PROFILE TUBES.

3.1 INTRODUCTION :

As has already been discussed, the film of condensate formed over the surface of the condenser tubes, accounts for the major resistance to heat flow in the process of film-condensation over horizontal tubes. The average unit conductance for heat transfer on the steam-side, in its most general form, associated with this process, is given by the expression (2.95), i.e.

$$\bar{h}_s = (\text{HTCF}) \times (0.728) \times \left[1+0.2\,\frac{C_{p_l}(T_{sat.}-T_w)}{h_{fg}}(n-1)\right] \times \left[\frac{\rho_l^2\, g\, k_l^3\, h'_{fg}}{n\, d_o\, \mu_l (T_{sat.}-T_w)}\right]^{1/4} \qquad (3.1)$$

where,

'HTCF' is a heat transfer coefficient factor, as explained in Chapter-II and given by expression (2.96) for different tube-profiles.

In expression (3.1), 'd_o' stands for the outer diameter of the tubes if they are circular and for polar profile

tubes, 'd_o' represents the outer diameter of a circular tube, which has been moulded to generate the polar profile. In either case, however, 'd_o' stands for the outer diameter of a circular profile.

Relation (3.1) indicates that the average steam-side heat transfer coefficient is inversely proportional to $(d_o)^{1/4}$. Reduction in 'd_o' , therefore, results in an increase in '$\bar{h}_s$' .

The water-side heat transfer coefficient is given by [4] ,

$$h_w = \left(\frac{k_w}{HMD}\right) \times (0.023) \times (Re)^{0.8} \times (Pr)^{0.4} \qquad (3.2)$$

'HMD' in expression (3.2) is the hydraulic mean diameter of the tube-profile and is given by the relation (2.58), i.e.,

$$HMD = \frac{4 \times AF \times (A_{cn})_c}{PER} \qquad (3.3)$$

where,

'AF' is an area factor, which is defined in Chapter-II and is given by expression (2.98), for different tube-profiles.

Since, '$(A_{cn})_c = \frac{\pi}{4} d_i^2$' and '$PER = \pi d_i$' the expression (3.3) for 'HMD' reduces to ,

$$HMD = (AF) \times d_i \qquad (3.4)$$

where,

'd_i' represents the inner diameter of the circular-profile tubes and in case of the polar profile-tubes, 'd_i' stands for the inside diameter of a corresponding circular profile, that has been moulded to generate the polar profile. In both cases, therefore, 'd_i' represents the inner diameter of a circular profile.

Using relations (3.4) and (3.2), the expression for 'h_w' becomes :

$$h_w = \left(\frac{1}{AF}\right) \times \left(\frac{k_w}{d_i}\right) \times (0.023) \times (Re)^{0.8} \times (Pr)^{0.4} \tag{3.5}$$

where

$$Re = \frac{(\rho_w\ V_w^{\circ}) \times HMD}{\mu_w}$$

or

$$Re = (AF) \times \frac{\rho_w\ V_w^{\circ}\ d_i}{\mu_w} \tag{3.6}$$

and

$$Pr = \frac{\mu_w\ C_{p_w}}{k_w} \tag{3.7}$$

The flow velocity of cooling water through the tubes 'V_w°' in expression (3.6) is given by,

$$V_w^{\circ} = \frac{\dot{V}}{A_{cn}} \tag{3.8}$$

where,

'$\dot{V}$' is the volume flow rate of cooling water and 'A_{cn}' denotes the area of cross section of the tubes. 'A_{cn}' is obtained from the relation (2.97), i.e.,

$$A_{cn} = (AF) \times (A_{cn})_c$$

or,

$$A_{cn} = (AF) \times \left(\frac{\pi}{4} d_i^2 \right) \tag{3.9}$$

Using (3.9), the expression (3.8) for 'V_w' reduces to ,

$$V_w = \left(\frac{1}{AF} \right) \times \frac{\dot{V}}{\left(\frac{\pi}{4} \cdot d_i^2 \right)} \tag{3.10}$$

Substituting (3.10) in (3.6), the expression for 'Re' becomes :

$$Re = \frac{4 \rho_w \dot{V}}{\pi d_i \mu_w} \tag{3.11}$$

Substituting the expression (3.11) for 'Re' and expression (3.7) for 'Pr' in (3.5) and simplifying we get,

$$h_w = \left(\frac{1}{AF} \right) \times \left(\frac{k_w}{d_i^{1.8}} \right) \times (0.023) \times \left(\frac{4 \rho_w \dot{V}}{\pi \mu_w} \right)^{0.8} \times \left(\frac{\mu_w Cp_w}{k_w} \right)^{0.4} \tag{3.12}$$

Expression (3.12), shows that the water-side heat transfer coefficient, i.e., 'h_w' is inversely proportional to $(d_i)^{1.8}$. For the same thickness of the tubes, a reduction in

the outer diameter requires a proportionate decrease in the inner diameter, which, in turn, results in an augmented water-side heat transfer coefficient.

Out of the remaining thermal resistances in the path of heat transfer from steam to the cooling water, the major contribution is rendered by the fouling of tubes , which is mainly a function of water quality and, hence, is independent of tube-dimensions. The tube-wall thermal resistance is very small in comparison with other resistances, encountered in the process. Thus, it can be safely neglected in the analysis, for all practical purposes.

Hence, any reduction in the tube diameter results in an increased overall heat transfer coefficient.

For a given condenser heat-duty, the flow rate of cooling water through the tubes of the condenser, before and after any change in diameter of the tubes, remains unaltered for a fixed temperature rise of cooling water. But in order to maintain the same flow rate with reduced inner diameter of tubes, requires a higher velocity of flow through the tubes. Since, the pressure drop in the tubes is proportional to the square of flow-velocity, the power required for pumping cooling water through the tubes is inreased. The pressure drop through the tubes is given by [4] ,

$$\Delta p = \frac{f \; L \; V_w^2 \; \rho_w}{2 \times HMD} \qquad (3.13)$$

Using expression (3.4) for 'HMD', the relation (3.13) for 'Δp' becomes :

$$\Delta p = \frac{f\ L\ V_w^2\ \rho_w}{2 \times (AF) \times d_i} \tag{3.14}$$

Substituting relation (3.10) for 'V_w' in (3.14), the expression for 'Δp' reduces to ,

$$\Delta p = \frac{8\ f\ L\ \dot{V}^2\ \rho_w}{(AF)^3 \times \pi^2 \times (d_i)^5} \tag{3.15}$$

It is evident from expression (3.15) that any decrease in the inner diameter of the condenser tubes results in a significantly high pressure drop through the tubes. This is because 'Δp' is inversely proportional to $(d_i)^5$. Hence, the power requirement for pumping cooling water through the condenser tubes is greatly enhanced.

To sum up, a decrease in the external diameter of the condenser tubes, resulting in an increase in the overall heat transfer coefficient, requires less surface area for causing a given heat transfer rate in the condenser. Hence, the amount of tube material, required for the purpose is reduced, giving rise to a reduction in the first-cost of the condenser. But the consequent increase in the energy requirement for pumping entails an increase in the energy-cost which is a running cost, associated with the analysis. Against these two conflicting requirements,

namely, the decreased first-cost and the increased running cost, the optimum tube diameter based on minimum total cost has to be determined. Since, the energy cost is a running cost, it varies with time and a futuristic approach should be made to account for this changing cost element with time over the total service-life of the condenser. The 'present worth method' is employed to perform this cost analysis in the ensuing sections of this chapter.

The following three sections, (3.2) to Section (3.4) prepare the ground for carrying out the detailed cost analysis.

3.2 DETERMINATION OF THE OVERALL HEAT TRANSFER COEFFICIENT 'U_o' AND THE THERMAL RESISTANCE 'R_m' DUE TO CONDENSER TUBE-FOULING FOR CIRCULAR AND POLAR PROFILE TUBES :

The following three factors contribute to the overall heat transfer coefficient.

(a) Steam-side thermal resistance, represented by 'R_s'.

(b) Water-side thermal resistance, denoted by 'R_w'.

and

(c) Miscellaneous thermal resistance which is attributed to the fouling of the condenser tubes and is represented by 'R_m'.

To determine 'R_s', '$\bar{h}_s$' is first evaluated by using expression (3.1), i.e.,

$$\bar{h}_s = (HTCF) x (0.728) x \left[1 + 0.2 \frac{C_{p_l}(T_{sat.} - T_w)}{h_{fg}} (n-1)\right] x \left[\frac{\rho_l^2 \, g \, h_{fg} \, k_l^3}{n \, d_o \, \mu_l (T_{sat.} - T_w)}\right]^{1/4}$$

'R_s' is then obtained with the help of the following relation :

$$R_s = \frac{1}{\bar{h}_s} \tag{3.16}$$

For obtaining 'R_w', 'h_w' is first determined with the help of the relation (3.12), i.e.,

$$h_w = \left(\frac{1}{AF}\right) x \left(\frac{k_w}{d_i^{1.8}}\right) x \, (0.023) x \left(\frac{4\rho_w \dot{V}}{\pi \mu_w}\right)^{0.8} x \left(\frac{\mu_w C_{p_w}}{k_w}\right)^{0.4}$$

'R_w' is then evaluated, using the following expression :

$$R_w = \left(\frac{A_o}{h_w \cdot A_i}\right) \tag{3.17}$$

where 'A_o' and 'A_i' stand for the 'outer surface area' and 'inner surface area' of the condenser tubes, respectively. They are determined as following :

For a condenser tube having

length = L m.

outer diameter= d_o m.

and inner diameter = d_i m. ,

'A_o' is given by,

$$A_o = \pi\, d_o\, L \tag{3.18}$$

and 'A_i' is given by,

$$A_i = \pi\, d_i\, L \tag{3.19}$$

Using (3.18) and (3.19), the expression (3.17) reduces to,

$$R_w = \frac{d_o}{d_i} \times \frac{1}{h_w} \tag{3.20}$$

The rate of heat transfer in the condenser is expressed as :

$$\dot{Q} = \dot{V} \times \rho_w \times C_{p_w} \times (T_o - T_i) \tag{3.21}$$

where,

'$\dot{V}$' is the volume flow rate of cooling water through the condenser-tubes.

'T_i' and 'T_o' represent the inlet and outlet temperatures of cooling water for the condenser, respectively.

'$\dot{Q}$' and 'U_o' are related through the following expression :

$$\dot{Q} = (U_o\, A_o) \times LMTD \tag{3.22}$$

where,

'LMTD' is the log mean temperature difference explained in Appendix-1.

Using (3.22), we get,

$$U_o = \frac{\dot{Q}}{A_o \times LMTD} \tag{3.23}$$

The expression which relates the sum of all the thermal resistances and 'U_o' is given by [4],

$$U_o = \frac{1}{\Sigma R} \tag{3.24}$$

where,

$$\Sigma R = R_s + R_w + R_m \tag{3.25}$$

Using (3.24), we get the expression for 'ΣR' as :

$$\Sigma R = \frac{1}{U_o} \tag{3.26}$$

Since, 'U_o' is already determined by using (3.23), 'ΣR' can be evaluated from relation (3.26). 'R_m' is then obtained by rearranging the terms in expression (3.25) as :

$$R_m = \Sigma R - R_s - R_w \tag{3.27}$$

As all the terms of the right hand side of (3.27) are already evaluated, 'R_m' can easily be determined.

For circular profile tubes, using 'HTCF =1' and 'AF=1' in expressions for '$\bar{h}_s$' and 'h_w', respectively, the miscellaneous thermal resistance 'R_m' is found to be 0.7281×10^{-3} m^2 $^oC/W$ when the total thermal resistance, i.e. 'ΣR' is obtained to

be 1.0187 x 10^{-3} m^2 $^oC/W$. Data for various parameters for the case study given in Appendix-1 have been used for the above calculations.

3.3 DETERMINATION OF A FUNCTIONAL RELATIONSHIP BETWEEN THE COST OF PRODUCTION OF ENERGY AND TIME :

'Appendix-10' gives the cost of production of energy at 'Panki Thermal Power Station, Kanpur' for the last seven consecutive years i.e. from the year 1979-80 to 1985-86. Considering the energy cost per unit as the dependent variable and year of observation as independent variable, a polynomial relation is obtained between the two variables. A 'computer programme No.7', listed in 'Appendix-11' has been employed to obtain this polynomial relationship by using the method of least squares . The polynomial, so obtained is given by,

$$CEPU\ (t) = 31.7742 + 0.5632\ t + 0.7124\ t^2 \qquad (3.28)$$

where,

'CEPU' represents the cost of production of energy per unit (KW. hr.) in Paise .

't' stands for the time-element given by the year of observation column in 'Appendix-10'. Substitution of t=1,2,......7 in expression (3.28) yields the 'CEPU' for the years 1979-80, 1980-81,,.......1985-86, respectively.

The above functional relationship is extrapolated to predict the energy cost per unit for the subsequent years. For example, substitution of t= 8,9,10,...... gives the 'CEPU' for years 1986-87, 1987-88, 1988-89......., respectively

This expression (3.28) will be employed in Section (3.5 for the determination of the running cost-element, involved in the cost analysis.

3.4 THE PRESENT WORTH METHOD :

A brief description of the present worth method, as employed in the cost analysis is presented in this section.

The total cost, involved in any cost analysis broadly comprises of the following two categories :

(a) First cost, denoted by 'FC'.

and

(b) Running cost, represented by 'RC'.

Since 'RC' is a time-dependent cost element, its effect is entirely different than 'FC' on the determination of the total cost. In any comparative cost analysis procedure, the present worth method acts as an elegant technique to account for the time-dependent and varying cost element, i.e., 'RC'. This method determines the present worth of the running cost, which is to be actually incurred at a future time

The following example explains the discussion in detail :

Suppose the total investment on an equipment for its procurement is 'FC' , i.e., the fixed cost.

Let the time-dependent running cost-element, associated with the operation of the equipment be denoted by 'RC' and 'l' be the life of the equipment in years.

'RC', being time-dependent, is denoted with a suffix. At the time of procurement of the equipment, i.e. at time, $t = 0$, 'RC' is represented as 'RC_o' , the magnitude of which will obviously be zero.

At the end of the 1st, 2nd, 3rd,........'l'th years, RC is represented as 'RC_1', 'RC_2', 'RC_3',......... 'RC_l'.

It is required to find the worth of the amounts 'RC_1', 'RC_2',.........'RC_l', which are to be spent at the end of 1,2,........ l years, respectively, from the date of procurement of the equipment, at time $t=0$. In other words, the present worth of the above amounts, which will be spent in a future period of time, is to be determined.

Let the present worth of 'RC_1' be denoted as 'PW_1'. This means that if 'PW_1' amount of money is deposited in the bank at time $t=0$, then at the end of one year, this amount 'PW_1', alongwith its interest, will yield an amount 'RC_1'.

Initial amount = PW_1

Suppose the annual rate of interest on bank-deposits is 'IR' in percentage.

$\therefore$ Total interest, earned at the end of one year $= PW_1 \times \frac{IR}{100}$

Hence, total amount received from the bank at the end of one year $= PW_1 + PW_1 \times \frac{IR}{100} = PW_1 \left(1 + \frac{IR}{100}\right)$

$$\therefore \quad PW_1 \left(1 + \frac{IR}{100} \right) = RC_1$$

$$\text{or,} \quad PW_1 = \frac{RC_1}{\left(1 + \frac{IR}{100}\right)} \tag{3.29}$$

Replacing ' $\frac{IR}{100}$ ' by ' X ', the expression (3.29) reduces to,

$$PW_1 = \frac{RC_1}{(1 + X)} \tag{3.30}$$

Similarly, if ' PW_2 ' is the present worth of RC_2, at the end of two years, this amount alongwith the interest will yield an amount ' RC_2 '.

Total amount that ' PW_2 ' will generate at the end of one year $= PW_2 \left(1 + \frac{IR}{100}\right)$ (3.31)

Total interest generated at the end of second year $= PW_2 \left(1 + \frac{IR}{100}\right) \times \left(\frac{IR}{100}\right)$ (3.32)

Hence, total amount, which ' PW_2 ' will yield at the end of second year = (3.31) + (3.32)

$$= PW_2 \left(1 + \frac{IR}{100} \right)^2$$

$$= PW_2 (1 + X)^2 \tag{3.33}$$

But this amount is the same as 'RC_2' from the very definition of present worth.

$$\therefore \quad RC_2 = PW_2 \, (1+X)^2$$

or,

$$PW_2 = \frac{RC_2}{(1+X)^2} \tag{3.34}$$

Extending the same concept further, the following results can be obtained.

$$PW_3 = \frac{RC_3}{(1+X)^3}$$

$$\vdots$$

$$PW_1 = \frac{RC_1}{(1+X)^1}$$

A generalised expression for the determination of present worth can be obtained from the above discussion as following :

If,

RC_i = Running cost, incurred at the end of the 'i'th year.

PW_i = The corresponding present worth.

Then, PW_i is given as :

$$PW_i = \frac{RC_i}{(1+X)^i} \tag{3.35}$$

Hence, the present worth of the running cost, incurred over the entire life-span of the equipment is given by,

$$PW_T = \sum_{i=1}^{l} PW_i \qquad (3.36)$$

where,

'PW_T' represents the present worth of the total running cost.

The total cost 'TC', then, is expressed as :

$$TC = FC + PW_T \qquad (3.37)$$

The total cost is calculated, using expression (3.37) for different equipments or different versions of the same equipment and that alternative, which leads to the minimum total cost is selected as the optimum choice.

This, in brief, epitomises the present worth method for cost analysis.

With the discussions in Sections (3.2) to (3.4) the stage is now set to undertake the complete cost analysis. The details of this analysis are presented in the next section.

3.5 DETAILS OF THE COST-OPTIMISATION ANALYSIS FOR CIRCULAR AND POLAR PROFILE TUBES IN A CASE STUDY;

3.5.1 Circular Tubes :

The cost-optimisation analysis is carried out as following :

In the case-study, the outer diameter of the condenser is 22mm. and thickness of the tubes is 1mm., i.e.

d_o = outer diameter of tubes = 22 mm.

t = thickness of tubes = 1 mm.

The outer diameter of the tubes is then changed continuously and its effect on total cost of the condenser (fixed cost + running cost) is observed in order to obtain the optimum outer diameter for minimum total cost. The corresponding other optimum dimensions can then be determined.

Hence, 'd_o' is a known quantity, initially.

For a fixed thickness of tubes, the inside diameter 'd_i' is then given by ,

$$d_i = d_o - 2t \tag{3.38}$$

The tube-wall surface temperature 'T_w' has already been determined in Section (2.3). With the knowledge of 'd_o', 'd_i' and 'T_w', the overall heat transfer coefficient 'U_o' is evaluated by following the same procedure as explained in Section (3.2).

The heat transfer rate '$\dot{Q}$' in the condenser and the overall heat transfer coefficient 'U_o' are related by the expression,

$$\dot{Q} = (U_o A_o) \times LMTD \tag{3.39}$$

Where,

'A_o' is the outer surface area of the condenser tubes and 'LMTD' represents the log mean temperature difference. '$\dot{Q}$' and 'LMTD' are given in 'Appendix-1'.

Hence, from expression (3.39), the outer surface area for heat transfer will be given by,

$$A_o = \frac{\dot{Q}}{U_o \times LMTD} \qquad (3.40)$$

But, as given in expression (3.18),

$$A_o = \pi \, d_o \, L$$

Hence, the length of the condenser tubes is given by,

$$L = \frac{A_o}{\pi \, d_o} \qquad (3.41)$$

The volume of the tube-material is then given as :

$$\forall_m = \frac{\pi}{4} (d_o^2 - d_i^2) \times L \qquad (3.42)$$

Where,

$\forall_m$ = Volume of tube material

The mass of tube-material used, is then expressed by,

$$M_m = \forall_m \times \rho_m \qquad (3.43)$$

Where,

ρ_m = Mass-density of the material for tubes

M_m = Mass of tube-material

The total cost, incurred in obtaining the tube-material for the condenser, i.e. 'C_m' is given by,

$$C_m = M_m \times C_{mt1} \tag{3.44}$$

Where,

C_{mt1} = Cost of tube-material in Rs per kg.

Since, the above analysis is carried out considering only two tubes (one for inlet and the other for the outlet of cooling water), the total cost of the tube-material is calculated by taking into account all the 13000 tubes of the condensing unit. Hence, the total material cost for the tubes is given by,

$$TC_m = C_m \times (13000/2) \tag{3.45}$$

Where,

TC_m = Total cost of the condenser tube-material.

The other element of cost, involved in the analysis is the energy cost for pumping cooling water through the condenser-tubes. The details of calculations, involved in determining this running cost element are as following :

The pressure drop for water, circulating through the tubes, is determined by using expression (3.14), i.e.,

$$\Delta p = \frac{f\ L\ V_w^2\ \rho_w}{2 \times (AF) \times d_i}$$

'AF' in the above expression is taken as unity for circular profile tubes as given in expression (2.98).

The only unknown in the expression for 'Δp' is 'f', which denotes the friction coefficient. If the flow of water through the tubes is laminar, no difficulty arises in calculating 'f', because it is then given by the simple expression [6],

$$f = \frac{64.0}{R_e} \tag{3.46}$$

Actual calculations, however, show that the flow Reynold's number is very large (of the order of 10^4 to 10^5) and hence the flow is invariably turbulent. Several simplified correlations have been developed to obtain 'f' for turbulent flow, but the most commonly used engineering practice is to use the coolebrook's equation [6], which forms the basis for drawing the Moody's diagram. This equation is given by,

$$\frac{1}{\sqrt{f}} = -2.0 \log_{10} \left(\frac{\varepsilon/d_i}{3.7} + \frac{2.51}{R_e \sqrt{f}} \right) \tag{3.47}$$

In the above expression 'ε' denotes the roughness coefficient for the tube-material. Values for different materials have been given in 'Appendix-12'.

Since, equation (3.47) is implicit in the variable 'f', a 'trial and error' method is needed to solve it for 'f'.

The most logical way of solving this problem is by using the 'Newton-Raphson's numerical technique', which is a very fast-converging method.

In the present analysis, subroutine <u>FRIXON</u> of 'computer programme No.8', given in 'Appendix-13', calculates 'f' by 'Newton-Raphson's numerical technique'.

Since all the parameters on the right hand side of the expression for 'Δp' are now determined, the pressure drop 'Δp' through the tubes of the condenser can be calculated.

The power, required for pumping cooling water through the condenser tubes is then given as :

$$P_w = \frac{\Delta p \times \dot{V}}{\eta_o} \quad [\,W\,] \tag{3.48}$$

Where,

'$\dot{V}$' represents the volume flow rate of cooling water, given in 'Appendix-1' and

'η_o' stands for the overall efficiency of the pump, which is assumed on the basis of practically encountered value to be <u>0.78</u> [4].

As has already been mentioned, the analysis is carried out for two tubes only. For the complete condenser, having 13000 tubes, the total power requirement for pumping is given by,

$$TP_w = P_w \times (13000/2) \quad [\,W\,] \tag{3.49}$$

As the condenser is continuously running (24 hrs. a day the electrical energy, consumed for pumping per year is expressed by,

$$E_p = \frac{TP_w \times 24 \times 365}{1000} \quad KW.\ hr. \qquad (3.50)$$

The cost of energy which is charged from a consumer towards the electricity bill is mainly a matter of Government poli and hence, it does not really reflect the actual cost of energy. The cost of production of energy, on the other hand, is more representative of its actual cost and therefore, is used in the present analysis. 'Appendix-10' gives the data for the cost of production of energy for the last seven consecutive years from 1979-80 to 1985-86 , collected from the 'PANKI THERMAL POWER STATION, KANPUR'.

Suppose, the year for the procurement of a new condenser is considered as 1986-87.

The approximate life of the unit is 25 years (Appendix 1).

Cost of energy, to be incurred at the end of 1986-87 is given by,

$$CE_1 = E_p \times CEN_1$$

Where,

CEN_1 = Cost of electrical energy per unit (KW.hr.) for the year 1986-87 (1st year)

and 'E_p' has been determined from relation (3.50)
Similarly, for the subsequent years,

$$CE_2 = E_p \times CEN_2$$

$$\vdots$$

$$CE_{25} = E_p \times CEN_{25}$$

In all these above expressions, CEN_1, CEN_2...... CEN_{25} are obtained by replacing 't' in expression (3.28) by 8,9,........ 32 respectively. The expression (3.28) is given as:

$$CEPU\ (t) = 31.7742 + 0.5632\ t + 0.7124\ t^2$$

Using the 'present worth method', discussed in Section (3.4), the following calculations are made.

With the help of relation (3.35), the present worth of the energy cost, which is to be incurred at the end of 1st year is given by,

$$PWCE_1 = \frac{CE_1}{(1+X)^1}$$

Where,

PWCE = Present worth of energy cost

X = Rate of interest on bank-deposits, assumed as <u>0.1</u>.

Suffix '1' indicates the end of 1st year.

Similarly,

$$PWCE_2 = \frac{CE_2}{(1+X)^2}$$

.
.
.
.

$$PWCE_{25} = \frac{CE_{25}}{(1+X)^{25}}$$

Therefore, the total cost of energy, incurred over the entire life of the condenser, as represented by the present worth method is given by the following expression :

$$TCE_p = \sum_{i=1}^{25} PWCE_i/100 \qquad (3.51)$$

In the above expression, the factor '100' is used to convert the cost from 'paise' to 'Rs'. 'TCE_p' represents the total cost of energy for pumping.

The total cost, involved in the process is then obtained by adding the material cost and the energy cost and is represented by,

$$TC = TC_m + TCE_p \qquad (3.52)$$

where,

TC = Total cost

and 'TC_m' is already calculated by using expression (3.45).

A 'computer programme No.8', listed in 'Appendix-13' calculates the total cost for different tube diameters and the

results are presented in 'Table 3.1' and 'Table 3.2'. In this tabular representation of the results of the cost analysis, 'ε=0.15 mm.' implies that the roughness coefficient of the condenser tube-material is assumed to be equal to that of 'galvanised iron' and 'ε=0.046 mm.' assumes this coefficient to be equal to that of 'commercial steel'. These assumptions are made because of the absence of data for 'ε' for the condenser tube-material, which is 'Aluminium Brass.'

The optimum tube dimensions and related results of the cost analysis are then calculated by using the following procedure :

At present, tubes of 22 mm. outer diameter are used in the condenser, taken for the case-study. Hence, all the parameters, given in 'Tables 3.1 and 3.2' are read against the row, denoted by 'DO=22'. These parameters are represented with suffix 'e' attached to them, which denotes existing .

When the total cost, denoted by 'CT' in the above tables, changes its nature from decreasing to increasing direction, this point of inflexion shows the optimum point. The optimum parameters of the cost analysis are then read against this point and suffix 'op' is attached to these parameters to denote optimum .

Total service life of the condenser =25 years approximately (Appendix-1).

Nomenclature for the various parameters is given as following :

(1) d_{op} = optimum outer diameter of condenser tubes

(2) SEECY = Saving in electrical energy consumption per year

(3) PSEEC = Percentage Saving in electrical energy consumption

(4) SECT = Saving in energy cost over 25 years

(5) ASEC = Annual saving in energy cost

(6) PSEC = Percentage saving in energy cost

(7) LMC = Loss in material cost

(8) PLMC = Percentage loss in material cost

(9) STCT = Saving in total cost over 25 years

(10) STCY = Saving in total cost per year

(11) PSTC = Percentage saving in total cost

The savings, indicated in the above parameters are for the optimum parameters over the existing parameters.

The expressions for evaluating the parameters (2) to (11) are given as :

$$SEECY = (ENCY_e - ENCY_{op}) \times 6500^{*} \quad (3.53)$$

$$PSEEC = (ENCY_e - ENCY_{op})/ENCY_e \times 100 \quad (3.54)$$

$$SECT = (CENT_e - CENT_{op}) \times 6500 \quad (3.55)$$

$$ASEC = SECT/25 \quad (3.56)$$

---

* The factor 6500 is used due to the following reason. The results of the cost analysis, presented in the 'Tables 3.1 to 3.6' are on the basis of two tubes in the condenser. Hence, to take into account all 13000 tubes of the condenser, all parameters of the tables are to be multiplied by (13000/2) or 6500.

$$PSEC = (CENT_e - CENT_{op})/CENT_e \times 100 \qquad (3.57)$$

$$LMC = (CMT_{op} - CMT_e) \times 6500 \qquad (3.58)$$

$$PLMC = (CMT_{op} - CMT_e)/CMT_e \times 100 \qquad (3.59)$$

$$STCT = (CT_e - CT_{op}) \times 6500 \qquad (3.60)$$

$$STCY = STCT/25 \qquad (3.61)$$

$$PSTC = (CT_e - CT_{op})/CT_e \times 100 \qquad (3.62)$$

In all the above expressions from (3.53) to (3.62), the magnitudes of the parameters of the right hand sides are read from the Tables 3.1 and 3.2. These parameters are also explained in the nomenclature of these tables. The results, calculated by the above procedure are given as following :

(a) circular profile, $\varepsilon = 0.15$ mm [Table 3.1]

$d_{op} = 52$ mm.

SEECY = 2887913 KW.hr., PSEEC = 99.4 %, SECT = 263573000 Rs./-
ASEC = 10542942 Rs/-, PSEC = 99.4 %, LMC = 4035070 Rs./-
PLMC = 56,43 %, STCT=259488000 Rs/-, STCY= 10379556 Rs./-
PSTC = 95.25 %

(b) circular profile, $\varepsilon = 0.046$ mm [Table 3.2]

$d_{op} = 51$ mm.

SEECY = 2191810 KW.hr., PSEEC= 99.2 %, SECT= 200041000 Rs./-
ASEC = 8001669 Rs./-, PSEC= 99.2 %, LMC = 3867565 Rs./-
PLMC = 54.1 %, STCT=196170000 Rs/-, STCY=7846802 Rs./-
PSTC = 93.9 %.

.5.2 Polar Profile Tubes :

In the expressions (3.1), (3.12) and (3.14), which ave been used in the cost analysis for circular profile tubes, AF' and 'HTCF' are replaced by the following values for the olar profiles:

Polar profile	AF	HTCF
$r = a\theta$	0.9758	1.0942
$r = ae^{\theta}$	0.8558	1.5713

The above values of 'AF' and 'HTCF' have been already valuated in Section (2.5.4).

The rest of the procedure is entirely the same as as been adopted for the circular profile tubes in Section (3.5). ithout repeating the procedure, therefore, only the final esults are presented.

(a) Profile, '$r=a\theta$', ε=0.15 mm. [Table 3.3]

d_{op} = 52 mm.

SEECY= 2991925 KW.hr., PSEEC= 99.4 % , SECT = 273066000 Rs./-

ASEC = 10922657 Rs./-, PSEC = 99.4 % , LMC = 3907670 Rs./-

PLMC = 54.7 % , STCT = 269054000Rs./-, STCY=10762190 Rs./-

PSTC = 95.5 %

(b) Profile, '$r=a\theta$', ε= 0.046 mm. [Table 3.4]

d_{op} = 51 mm.

SEECY = 2271101 KW.hr , PSEEC=99.2 % , SECT = 207278000 Rs./-

ASEC = 8291137 Rs./- , PSEC =99.2 % , LMC = 3740880 Rs./-

PLMC = 52.3 % , STCT = 203433000 Rs./-, STCY = 8137342 Rs./-

PSTC = 94.2 %

(c) Profile, '$r=ae^{\theta}$', ε= 0.15 mm. [Table 3.5]

d_{op} = 54 mm.

SEECY = 3723774 KW.hr., PSEC = 99.5 % , SECT = 339860000 Rs./-

ASEC = 13594427 Rs./-, PSEC = 99.5 % , LMC = 4260295 Rs./-

PLMC = 63.4 % , STCT =335600000 Rs./-, STCY=13424015 Rs./-

PSTC = 96.3 %

(d) Profile, '$r=ae^{\theta}$', ε = 0.046 mm. [Table 3.6]

d_{op} = 53 mm.

SEECY = 2829085 KW.hr., PSEEC = 99.3 % , SECT = 258204000 Rs./-

ASEC = 10328177 Rs./-, PSEC = 99.3 % , LMC =4090905 Rs./-

PLMC = 60.8 % , STCT = 254113000 Rs./-, STCY =10164541 Rs./-

PSTC = 95.26 %

Another aspect of the results of cost analysis [Tables 3.1 to 3.6], which is used to find the saving in tube-material by using polar profile tubes in the condenser over the corresponding circular profile tubes, is presented as following :

The condenser taken for analysis in the case-study, uses circular tubes of 22 mm. outer diameter. If these tubes are

changed to the polar profiles, then there is a reduction in the tube-material, required for effecting the same heat transfer rate in the condenser. This causes a decrease in the space-requirement of the condenser and also, there is a reduction in the first cost.

The actual magnitudes of the above results are obtained for polar profiles '$r= a\theta$' and '$r=ae^{\theta}$' respectively.

PROFILE '$r = a\theta$'

Saving in material = (MTM from Table (3.1) - MTM from Table (3.3) x 6500

= (10-9.855)x 6500 = 942.5 kg.

where, 'MTM' represents the mass of tube-material and is read against the row, given by 'DO=22' in the tables.

% saving in material = $\frac{10-9.855}{10}$ x 100 = 1.45 %

Saving in first cost of the condenser= Saving in material x cost of material

= 942.5 x 110

= 103675 Rs./-

The savings, indicated above are for the polar profile, '$r= a\theta$' over the corresponding circular profile.

PROFILE '$r = ae^{\Theta}$'

Saving in material = (MTM from Table (3.1)- MTM from Table (3.5) x 6500

$= (10-9.406) \times 6500 = \underline{3861}$ kg.

Where, 'MTM' denotes the mass of tube-material and is read against the row, given by 'DO = 22' in the tables.

$$\% \text{ Saving in material} = \frac{10-9.406}{10} \times 100 = \underline{5.94}\ \%$$

Saving in first cost of the condenser = Saving in material x cost of material

$= 3861 \times 110$

$= \underline{424710}$ Rs./-

The savings, indicated above are for the polar profile, '$r = ae^{\Theta}$' over the corresponding circular profile.

Nomenclature for Tables (3.1)- (3.6)

DO	:	Outer diameter of tubes [mm.]
FF	:	Friction factor [dimensionless]
ENCY	:	Energy, consumed per year [Kw.hr.]
CENT	:	Total energy cost over the entire service life of the condenser [Rs.]
CMT	:	Total cost of tube material [Rs.]
CT	:	Total cost, involved, for the entire service life of the condenser [Rs.]
MTM	:	Mass of tube material [Kg.]
L	:	Length of tubes [m]
ε	:	Roughness coefficient of tube-material [mm.]

Note : This analysis is made on the basis of one set of two tubes (one each for entrance and exit of cooling water) of the condenser. There are 6500 such sets in one complete condensing unit. Results for the complete condensing unit, therefore, are obtained by multiplying these results by 6500.

Table 3.2 Results of cost analysis

(Circular profile, ε = 0.046)

DO	FF	ENCY	CENP	CMT	CT	MTM	L
20	0.0271	621.1309	56689.26	1067.73	57757.00	9.707	16.09
22	0.0268	340.0339	31034.16	1099.37	32133.53	9.994	14.99
25	0.0264	154.5171	14102.45	1148.48	15250.93	10.441	13.70
30	0.0262	52.1206	4756.94	1235.88	5992.82	11.235	12.20
35	0.0262	21.5328	1965.25	1331.26	3296.51	12.102	11.21
40	0.0263	10.2708	937.40	1435.19	2372.59	13.047	10.54
45	0.0265	5.4496	497.37	1547.94	2045.32	14.072	10.07
50	0.0268	3.1374	286.34	1669.61	1955.95	15.178	9.76
51	0.0269	2.8324	258.51	1695.01	1953.52	15.409	9.71
52	0.0269	2.5633	233.95	1720.78	1954.73	15.643	9.66
53	0.0270	2.3252	212.22	1746.90	1959.12	15.881	9.62
54	0.0271	2.1140	192.94	1773.38	1966.32	16.122	9.58
55	0.0271	1.926	175.79	1800.21	1976.00	16.366	9.55

Table 3.3 Results of cost analysis

(Profile, 'r=aθ', ε =0.15)

DO	FF	ENCY	CENT	CMT	CT	MTM	L
20	0.0366	860.2922	78517.00	1052.79	79569.79	9.571	15.87
22	0.0356	463.2828	42282.81	1084.00	43366.81	9.855	14.78
25	0.0344	205.7027	18774.04	1132.51	19906.55	10.296	13.51
30	0.0330	67.0279	6117.50	1219.05	7336.55	11.082	12.04
35	0.0321	26.8843	2453.67	1313.68	3767.35	11.943	11.06
40	0.0315	12.5078	1141.56	1416.95	2558.51	12.881	10.40
45	0.0311	6.4998	593.22	1529.10	2122.33	13.901	9.95
50	0.0309	3.6778	335.67	1650.22	1985.89	15.002	9.64
51	0.0308	3.3100	302.10	1675.52	1977.63	15.232	9.60
52	0.0308	2.9867	272.59	1701.18	1973.77	15.465	9.55
53	0.0308	2.7015	246.56	1727.20	1973.77	15.702	9.51
54	0.0308	2.4493	223.54	1753.58	1977.12	15.942	9.47
55	0.0307	2.2256	203.13	1780.32	1983.45	1[illegible].185	9.44

Table 3.4 Results of cost analysis

(Profile, 'r=aθ', ε =0.046)

DO	FF	ENCY	CENT	CMT	CT	MTM	L
20	0.0271	643.5441	58734.87	1052.79	59787.65	9.571	15.87
22	0.0268	352.3460	32157.85	1084.00	33241.85	9.855	14.78
25	0.0264	160.1526	14616.78	1132.51	15749.29	10.296	13.51
30	0.0262	54.0518	4933.20	1219.05	6152.25	11.082	12.04
35	0.0262	22.3453	2039.42	1313.68	3353.09	11.943	11.06
40	0.0263	10.6659	973.46	1416.95	2390.41	12.881	10.40
45	0.0265	5.6631	516.87	1529.10	2045.97	13.901	9.95
50	0.0268	3.2625	297.77	1650.22	1947.99	15.002	9.64
51	0.0269	2.9458	268.86	1675.52	1944.38	15.232	9.60
52	0.0269	2.6663	243.35	1701.18	1944.53	15.465	9.55
53	0.0270	2.4189	220.78	1727.20	1947.98	15.702	9.51
54	0.0271	2.1994	200.74	1753.58	1954.33	15.942	9.47
55	0.0271	2.0042	182.92	1780.32	1963.24	16.185	9.44

Table 3.5 Results of cost analysis

(profile, 'r=ae$^{\theta}$', ε =0.15)

DO	FF	ENCY	CENT	CMT	CT	MTM	L
20	0.0366	1069.1927	97582.88	1004.85	98587.73	9.185	15.14
22	0.0356	575.9751	52567.99	1034.66	53602.65	9.406	14.11
25	0.0344	255.9381	23358.92	1081.28	24440.20	9.830	12.90
30	0.0330	83.5478	7625.23	1165.06	8790.29	10.591	11.50
35	0.0321	33.5836	3065.11	1257.29	4322.40	11.430	10.59
40	0.0315	15.6615	1429.39	1358.44	2787.83	12.349	9.97
45	0.0311	8.1581	744.58	1468.67	2213.24	13.352	9.56
50	0.0309	4.6269	422.29	1588.03	2010.32	14.437	9.28
51	0.0308	4.1661	380.24	1613.00	1993.24	14.664	9.24
52	0.0308	3.7608	343.24	1638.33	1981.58	14.894	9.20
53	0.0308	3.4032	310.61	1664.03	1974.64	15.128	9.16
54	0.0308	3.0868	281.73	1690.09	1971.82	15.364	9.13
55	0.0307	2.8061	256.11	1716.51	1972.62	15.848	9.08

Table 3.6 Results of cost analysis

(profile, '$r=ae^{\theta}$', ε =0.046)

DO	FF	ENCY	CENT	CMT	CT	MTM	L
20	0.0271	800.1975	73032.28	1004.85	74037.13	9.135	15.14
22	0.0268	438.2931	40002.06	1034.66	41036.72	9.406	14.11
25	0.0264	199.3901	18197.90	1081.28	19279.18	9.830	12.90
30	0.0262	67.4227	6153.52	1165.06	7318.58	10.591	11.50
35	0.0262	27.9352	2549.59	1257.29	3806.88	11.430	10.59
40	0.0263	13.3655	1219.84	1358.44	2578.28	12.349	9.97
45	0.0265	7.1132	649.21	1468.67	2117.88	13.352	9.56
50	0.0268	4.1073	374.87	1588.03	1962.90	14.437	9.28
51	0.0269	3.7101	338.62	1613.00	1951.62	14.664	9.24
52.	0.0269	3.3596	306.63	1638.33	1944.96	14.894	9.20
53	0.0270	3.0493	278.30	1664.03	1942.33	15.128	9.16
54	0.0271	2.7738	253.16	1690.09	1943.25	15.364	9.13
55	0.0271	2.5286	230.78	1716.51	1947.29	15.605	9.10

CHAPTER-IV

DISCUSSION OF RESULTS

A numerical technique to solve the problem of determining the condensation heat transfer coefficient for any shape of tube-profile in a condenser has been developed in this thesis. The heat transfer augmentation effects, obtained by using the polar profile tubes, given by polar curves '$r=a\theta$' and '$r=ae^{\theta}$', in comparison with circular profile tubes in a condenser, are presented. The corresponding reduction in the first cost of the equipment is also evaluated. The optimum condenser-tube dimensions are determined on the basis of minimum total cost of the equipment. The condensing unit of the 110 MW system of the 'Panki Thermal Power Station, Kanpur' has been taken as a case-study for the above analysis. Solutions for this specific case have been presented in the form of tables and figures (Tables 2.1 to 2.5, Tables 3.1 to 3.6, figures 2.2 to 2.5). Important results on the basis of these tabularly represented solutions have been obtained in Chapters II and III. In this Chapter, a detailed discussion of these results is presented as following :

For convenience of representation, the polar profile of condenser tubes, given by '$r=a\theta$' is represented as 'PPCT-1' and for the curve '$r=ae^{\theta}$', it is denoted by 'PPCT-2'.

The use of 'PPCT-1' in place of circular tubes in the condenser results in an increment of 9.42 % in the heat transfer rate [Chapter-II]. 7.22 % of this increase is due to the contribution of gravity force and the rest 2.20 % is because of the surface-tension force. Hence, the effect of gravity is predominant in causing the heat transfer augmentation. The use of 'PPCT-2' causes an increase of 57.13 % in heat transfer, out of which, 33.03 % contribution is due to gravity force and 24.10 % is the effect of surface-tension force. Therefore, both effects have almost even contributions for causing the heat transfer augmentation in case of 'PPCT-2'.

As has been pointed out in Section (3.2), the thermal-resistance across the heat-flow path in the condenser due to tube-fouling is found to be $\underline{0.7281 \times 10^{-3}\ m^2\ ^oC/W}$ when the total thermal-resistance is 1.0187×10^{-3}. Hence, fouling of condenser tubes, accounts for 71.4 % of the total thermal resistance, which is quite large. This warrants regular cleaning and maintenance of the tubes to keep this value of thermal-resistance equal to the usually accepted value in commercial practice (of the order of $0.176 \times 10^{-3}\ m^2\ ^oC/W$ [4]). This will ensure a high heat transfer in the condenser and hence, a smaller size condenser is required for a given heat-duty.

From the results, presented in Section (3.5.1), it is observed that using 'PPTC-1' and 'PPTC-2', respectively, instead

of the circular tubes in the condenser, result in a reduction in condenser-space requirement of 1.45 % and 5.94 % respectively. This, in addition to the discussion of the previous paragraph, establishes the fact that whever, in an industrial system using a condenser,space is a primary concern, 'PPTC-2' may be used in place of circular tubes in the condenser.

The most important results, however, are those, obtained from the cost-optimisation analysis in Section (3.5). Considerable saving in total cost of the condenser has been seen to accrue from the use of optimum tube-dimensions over the existing dimensions. For example, changing the outer diameter of the condenser tubes from 22mm. to the optimum diameter, which is found to be 52 mm. for ε=0.15 mm [Section 3.5], will cause an annual reduction in total cost, amounting to Rs.10379556/- for the entire service life of the condenser.

Moreover, using the optimum diameter of the condenser tubes, mentioned above (52 mm), also results in an annual saving in electrical energy, amounting to 2887913 KW.hr. over the existing tube-diameter (22mm.). This is a fabulous saving in electrical energy, which is a very precious commodity, specifically in the wake of the present energy-crisis. Hence, adoption of optimum tube-dimensions in the condensers of all the thermal power plants of our country will result in an incredibly large saving in energy, which can be diverted to many other energy-starved areas.

Similar results of the cost-optimisation analysis for 'PPTC-1' and 'PPTC-2' have been presented in Section (3.5.1).

An apparent demerit in the use of proposed polar profile tubes in place of circular tubes in the condenser seems to be that the manufacturing cost of these unconventional-shaped tubes may be very high and this may cause a large increase in the first cost of the equipment. But in all probability, if the manufacturing processes for these tubes are standardized, the cost, involved in their manufacturing will be comparable with that of the corresponding circular tubes and therefore, not much increase in the first cost of the condenser is expected.

APPENDIX -1

OPERATING DATA OF THE CONDENSER, USED FOR THE 110MW. UNIT AT THE PANKI THERMAL POWER STATION, KANPUR .

(a) Condenser pressure(Kgf/cm^2):

		Max.	Min.	Normal
(Vacuum)	:	0.93	0.80	0.85
(Absolute)	:	0.103	0.233	0.183

(b) Cooling water temperature
at the inlet to the condenser : $32^oC = T_i$

(c) Cooling water temperature
at the outlet from the condenser: $40^oC = T_o$

(d) Tube size :
(i) Outer diameter = 22mm. = d_o
(ii) Inner diameter = 20 mm.= d_i
(iii) Length = 7.5m. = L_t

(e) Material of Construction
of the condenser tubes : Aluminium Brass.

(f) Volume flow rate of cooling
water through the condenser : 15400 m^3/hr.

(g) No. of hours, the condenser
operates in a day : 24 hrs.

(h) Approximate life of the
condenser. : 25 yrs

(i) No. of tubes in the condenser : 13,000

(j) Arrangement of tubes : 2-tube-pass, shell and tube arrangement.

(k) Average number of tubes per vertical row. : 40

(l) Weight of one tube of 7.5 m. length. : 5 Kg.

(m) Cost of pipe material : Rs. 110 per Kg.

DATA, DERIVED FROM THOSE, SPECIFIED ABOVE :-

(n) Saturation temperature of steam in the condenser at normal pressure. : $T_{sat} = 57.76^{o}C$

(o) Latent heat of evaporation of steam at the normal pressure : $h_{fg} = 2364064$/J/Kg.

(p) Length of tube, to be considered for evaluating pressure drop : L = 15 m.

(q) The volume flow rate of cooling water through one set of tubes is calculated as follows :

Total volume flow rate of cooling water

$$= \dot{V}_T = 15,400 \text{ m}^3/\text{hr}.$$

Water flows in through one set of tubes and flows out of the condenser through another set of tubes. Hence, only half of the total number of tubes are to be considered for calculating the flow rate through one-pass.

$\therefore$ Volume flow rate of cooling water through one-pass (consisting of one tube from each set, above)

$$= \dot{V} = \frac{15400}{(13000/2)}$$

$$= 2.3692307 \ m^3/hr.$$

$$= \frac{2.3692307}{3600}$$

$$= 6.58119 \ x10^{-4} \ m^3/sec.$$

(r) The mass-density of the material of condenser tubes is calculated as follows :

A tube of 7.5m. length with 'd_o=22mm.' and 'd_i = 20mm.' weighs 5 Kg.

Volume of the above tube

$$= \frac{\pi}{4} (d_o^2 - d_i^2) x10^{-6} \ x \ 7.5$$

$$= 4.948 \ x \ 10^{-4} \ m^3$$

Mass-density of the tube material is then given by,

$$\rho_m = \frac{5}{4.948 \ x10^{-4}} \ Kg./m^3$$

$$= 10105 \ Kg./m^3$$

(s) The LMTD (log mean temperature difference) is evaluated as follows :

$$\text{LMTD} = \frac{(T_{sat.} - T_o) - (T_{sat.} - T_i)}{\log_e \left(\frac{T_{sat.} - T_o}{T_{sat.} - T_i} \right)}$$

$$= \frac{(57.76-40) - (57.76-32)}{\log_e \left(\frac{57.76-40}{57.76-32} \right)}$$

$$= \frac{-8}{\log_e \left(\frac{17.76}{25.76} \right)}$$

$$= 21.51 \ ^{o}C$$

(t) $\dot{Q}$ = Heat Transfer rate through 15m. length of the condenser $= \dot{m}_w \, C_{p_w} (T_o - T_i) = \rho_w \, \dot{V} \, C_{p_w} (T_o - T_i) = 1000 \times \dot{V}$

$\times \, 4.184 \, (40-32)$

$= \underline{21890 \text{ W}}$

APPENDIX -2

THERMOPHYSICAL PROPERTIES OF CONDENSATE(WATER) [3]

Nomenclature :

T : Temperature [^{o}C]

ρ : Mass-density [$Kg./m^3$]

C_p : Specific heat [$J/Kg.\ ^{o}C$]

k : Thermal conductivity [$W/m\ ^{o}C$]

μ : Dynamic Viscosity [$N.s/m^2$]

Interpretation :

To use the last column of the data (shown on the next page in a tabular form), the following procedure is adopted.

Let the value, read from this column for μ be 792.4 .

Then, the actual value of μ is

$$792.4 \times 10^{-6} \quad N.s/m^2$$

....Contd...

Appendix-2 (contd.)

T	ρ	C_p	k	$\mu \times 10^6$
30.0	995.70	4176.00	0.615	792.4
32.0	995.10	4175.60	0.619	763.4
35.0	994.10	4175.00	0.624	719.8
37.0	993.34	4175.00	0.628	695.1
40.0	992.20	4175.00	0.633	658.0
42.0	991.40	4175.40	0.636	636.9
45.0	990.20	4175.00	0.640	605.1
47.0	989.36	4176.80	0.643	585.1
50.0	988.10	4178.00	0.647	555.1
52.0	987.04	4178.96	0.649	540.8
55.0	985.46	4180.40	0.652	519.4
57.0	984.40	4181.36	0.654	505.1
60.0	982.82	4182.80	0.657	483.7

APPENDIX-3

```
C                      ----------------------
C                      COMPUTER PROGRAMME No.1
C                      ----------------------
C     ************************************************************
C     PROGRAMME TO EVALUATE THE FUNCTIONAL RELATIONSHIP
C     OF THERMAL CONDUCTIVITY(K),DYNAMIC VISCOSITY(MU),
C     SPECIFIC HEAT(CP) AND MASS DENSITY(RHO) WITH THE
C     VARIATION OF TEMPERATURE USING THE 'METHOD OF LEAST SQUARES'.
C     ************************************************************
      DIMENSION S(10,10),X(100),B(5),A(5),Y(100),CY(100),PD(100)
C     ************************************************************
C     NOMENCLATURE:---
C     M=NUMBER OF DATA POINTS
C     N=ORDER OF POLYNOMIAL TO BE FITTED THROUGH THE DATA POINTS
C     X=INDEPENDENT VARIABLE(TEMP.)
C     Y=DEPENDENT VARIABLE(RHO,MU,K OR CP)
C     CY=DEPENDENT VARIABLE CALCULATED FROM THE POLYNOMIAL-FIT.
C     PD=PERCENTAGE DEVIATION
C     SUM=CUMULATIVE 'PD'.
C     ************************************************************
      WRITE(49,31)
   31 FORMAT(////////////////)
      M=13
      N=2
C     ************************************************************
C     THE FOLLOWING FILE GIVES INPUT DATA TO THE PROGRAMME
C     ************************************************************
      READ(24,*)(X(I),Y(I),I=1,M)
C     ************************************************************
C     EVALUATION OF THE ELEMENTS OF THE MATRIX 'S' AND MATRIX 'B'.
C     ************************************************************
      SUMY=0.
      SUMXY=0.
      SUMXSY=0.
      SUMXCY=0.
      SUMX=0.
      SUMXSR=0.
      SUMXCB=0.
      SUMXFR=0.
      SUMXFV=0.
      SUMXSX=0.
      DO 10 I=1,M
      SUMY=SUMY+Y(I)
      SUMXY=SUMXY+X(I)*Y(I)
      SUMXSY=SUMXSY+(X(I)**2)*Y(I)
      SUMXCY=SUMXCY+(X(I)**3)*Y(I)
      SUMX=SUMX+X(I)
      SUMXSR=SUMXSR+X(I)**2
      SUMXCB=SUMXCB+X(I)**3
      SUMXFR=SUMXFR+X(I)**4
      SUMXFV=SUMXFV+X(I)**5
      SUMXSX=SUMXSX+X(I)**6
   10 CONTINUE
      S(1,1)=M
      S(1,2)=SUMX
      S(1,3)=SUMXSR
      S(1,4)=SUMXCB
      S(2,1)=SUMX
      S(2,2)=SUMXSR
      S(2,3)=SUMXCB
      S(2,4)=SUMXFR
      S(3,1)=SUMXSR
      S(3,2)=SUMXCB
      S(3,3)=SUMXFR
```

```
C     S(3,4)=SUMXFV
C     S(4,1)=SUMXCB
C     S(4,2)=SUMXFR
C     S(4,3)=SUMXFV
C     S(4,4)=SUMXSX
      B(1)=SUMY
      B(2)=SUMXY
      B(3)=SUMXSY
C     B(4)=SUMXCY
      N=N+1
      WRITE(21,*)((S(I,J),J=1,N),I=1,N),(B(I),I=1,N)
      CLOSE(UNIT=21,DEVICE='DSK')
      READ(21,*)((S(I,J),J=1,N),I=1,N),(B(I),I=1,N)
C     ******************************
C     INVERSION OF THE MATRIX 'S'.
C     ******************************
      I=1
      NX=N+1
      NY=2*N
      DO 80 J=NX,NY
      S(I,J)=1.
      I=I+1
 80   CONTINUE
      L=1
      K=2
 110  XM=S(L,L)
      DO 140 J=L,NY
      S(L,J)=S(L,J)/XM
 140  CONTINUE
      DO 190 I=K,N
      X1=S(I,L)
      DO 191 J=L,NY
      S(I,J)=S(I,J)-S(L,J)*X1
 191  CONTINUE
 190  CONTINUE
      L=L+1
      K=K+1
      IF(L-N)110,110,230
 230  L=N
 235  LZ=L-1
      DO 290 K=1,LZ
      I=L-K
      Y1=S(I,L)
      DO 291 J=L,NY
      S(I,J)=S(I,J)-S(L,J)*Y1
 291  CONTINUE
 290  CONTINUE
      L=L-1
      IF(L-1)320,320,235
 320  WRITE(23,*)((S(I,J),J=NX,NY),I=1,N)
      CLOSE(UNIT=23,DEVICE='DSK')
      READ(23,*)((S(I,J),J=NX,NY),I=1,N)
```

```
C      ***************************************************************
C      MULTIPLICATION OF THE INVERTED MATRIX OF 'S' WITH MATRIX 'B'.
C      ***************************************************************
       DO 70 I=1,N
       A(I)=0.
       DO 71 J=NX,NY
       A(I)=A(I)+S(I,J)*B(J-N)
71     CONTINUE
70     CONTINUE
C      *******************************************************************
C      THE ROW MATRIX 'A' GIVES THE COEFF.S OF THE FITTED POLYNOMIAL.
C      *******************************************************************
       WRITE(48,*)(A(I),I=1,(NY-N))
       CLOSE(UNIT=48,DEVICE='DSK')
       READ(48,*)(A(I),I=1,(NY-N))
C      *******************************************************************
C      CALCULATION OF THE DEVIATION BETWEEN THE ACTUAL VALUE AND
C      CALCULATED VALUE OF THE DEPENDENT VARIABLE.
C      *******************************************************************
       SUM=0.
       DO 27 J=1,M
       CY(J)=0.
       DO 28 I=1,N
       CY(J)=CY(J)+(A(I)*(X(J)**(I-1)))
28     CONTINUE
       PD(J)=((CY(J)-Y(J))/Y(J))*100.
       SUM=SUM+PD(J)
       WRITE(49,*)X(J),Y(J),CY(J),PD(J),SUM
27     CONTINUE
       STOP
       END

.type x
```

APPENDIX -4

POLYNOMIAL RELATIONS BETWEEN THE THERMOPHYSICAL PROPERTIES OF THE CONDENSATE(WATER)AND TEMPERATURE

1. $\rho = 1000.922 - (0.1757813 \times 10^{-1})T - (0.3875732 \times 10^{-2})T^2$

2. $C_p = 4196.031 - (1.010742)T + (0.1684570 \times 10^{-1})T^2$

3. $k = 0.5383759 + (0.3154755 \times 10^{-2})T - (0.1913682 \times 10^{-4})T^2$

4. $\mu = 1371.895 - (23.90112)T + (0.1526718)T^2$

Note : Values of thermophysical properties, listed in 'Appendix-2' and 'computer programme No.1', given in 'Appendix-3', have been used to develop the above corelations.

APPENDIX-5

COMPUTER PROGRAMME No.2

```
C          ************************************************************
C          PROGRAMME FOR OBTAINING THE TUBE-WALL SURFACE TEMPERATURE
C          BY ITERATIONS USING 'IF' LOOP.
C          ************************************************************
C          IL=INITIAL LENGTH
C          K=THERMAL CONDUCTIVITY OF LIQUID FILM
C          MU=DYNAMIC VISCOSITY OF LIQUID FILM
C          CP=SPECIFIC HEAT OF LIQUID FILM
C          RHO=MASS DENSITY OF LIQUID FILM
C          HFG=LATENT HEAT OF VAPOURISATION
C          HFGSC='HFG' CONSIDERING SUBCOOLING OF THE LIQUID FILM
C          DO=OUTER DIAMETER OF TUBE
C          AO=OUTER SURFACE AREA OF THE TUBE
C          HO=HEAT TRANSFER COEFF. OF CONDENSATION
C          CCFSC=CHEN'S CORRECTION FACTOR FOR SUBCOOLING OF CONDENSATE
C          ANT=AVERAGE NO. OF TUBES IN A VERTICAL ROW
C          TSAT=SATURATION TEMP.
C          TW=WALL TEMP. OF THE TUBE
C          TREF=REFERENCE TEMP.
C          QOR=ORIGINAL HEAT TRANSFER RATE IN THE CONDENSER
C          QNEW=NEW HEAT TRANSFER RATE AFTER ITERATION
C          ADG=ACCELERATION DUE TO GRAVITY
C          PCD=PERCENTAGE DEVIATION BETWEEN 'QOR' AND 'QNEW'
C          *******************************************************
C          FRHO,FK,FMU,FCP ARE FUNCTIONS DEPICTING THE VARIATION
C          WITH TEMP. OF 'RHO','K','MU AND 'CP' RESPECTIVELY.
C          *******************************************************
           REAL IL,MU,K
           FRHO(X)=1000.922-0.1757813E-01*X-0.3875732E-02*(X**2)
           FCP(X)=4196.031-1.010742*X+0.1684570E-01*(X**2)
           FK(X)=0.5383759+0.3154755E-02*X-0.1913682E-04*(X**2)
           FMU(X)=(1371.895-23.90112*X+0.1526718*(X**2))/(10.**6)
           WRITE(40,11)
      11   FORMAT(///////////////)
           HFG=2364064.0
           TSAT=57.76
           PI=3.14159
           ADG=9.81
           DO=0.022
           IL=15.0
           AO=PI*DO*IL
           QOR=21890.0
C          ************************************************************
C          THE INITIAL GUESS VALUE OF 'TW' IS GIVEN AND THE 'IF' LOOP
C          STARTS.THE TERMINATION CRITERION IS THAT 'PCD' IS LESS THAN
C          OR EQUAL TO 0.01 PERCENT.
C          ************************************************************
           TW=54.0
      28   TREF=TW+0.25*(TSAT-TW)
           MU=FMU(TREF)
           K=FK(TREF)
           CP=FCP(TREF)
           RHO=FRHO(TREF)
           HFGSC=(HFG)+((3./8.)*(CP)*(TSAT-TW))
           ANT=40.0
           CCFSC=1.0+(0.2*CP*(TSAT-TW)*(ANT-1.0))/HFG
           HOF1=(RHO**2)*(HFGSC)*(ADG)*(K**3)
```

```
      HOF2=(MU)*(TSAT-TW)*(ANT)*(DO)
      HOF=(HOF1/HOF2)**0.25
      HO=(0.728)*(CCFSC)*(HOF)
      QNEW=(HO)*(AO)*(TSAT-TW)
      PCD=(ABS(QOR-QNEW)/QOR)*100.0
      IF(PCD.LE..01)GO TO 29
      TW=TW+0.001
      GO TO 28
29    WRITE(40,9)TW,HO,QOR,QNEW,K,MU,RHO,CP,CCFSC,HFGSC
9     FORMAT(15X,'TW='F8.5//15X,'HO='F8.3//15X,'QOR='F8.2//15X,'QNEW='
     1F8.2//15X,'K='F8.6//15X,'MU='E13.7//15X,'RHO='F8.3//15X,'CP='
     1F8.3//15X,'CCFSC='F8.6//15X,'HFGSC='F8.0)
      STOP
      END
```

.k/f

APPENDIX-6

```
C             ------------------------
C             COMPUTER PROGRAMME No.3
C             ------------------------
C      ******************************************************
C      PROGRAMME TO OBTAIN THE HEAT TRANSFER COEFFICIENT
C      AT VARIOUS DISCRETE LOCATIONS ALONG THE PERIPHERY
C      OF THE CONDENSER TUBES OF CIRCULAR PROFILE.
C      ******************************************************
C      NOMENCLATURE:--
C      RMF=RADIUS MULTIPLYING FACTOR.
C      RAD=RADIUS OF THE CIRCULAR PROFILE.
C      A=CONSTANT 'A' IN THE POLAR EXPRESSION OF THE FORM
C       R=A*F(THETA) .
C      TREF=REFERENCE TEMPERATURE.
C      RHOL=DENSITY OF LIQUID.
C      KL=THERMAL CONDUCTIVITY OF LIQUID.
C      MUL=DYNAMIC VISCOSITY OF LIQUID.
C      SIGL=SURFACE TENSION OF THE LIQUID.
C      HFG=LATENT HEAT OF VAPOURISATION AT SATURATION PRESSURE.
C      HFGSC='HFG' CONSIDERING THE EFFECT OF SUBCOOLING OF THE
C      CONDENSATE.
C      CFSC=CHEN'S CORRECTION FACTOR FOR SUBCOOLING.
C      ANT=AVERAGE NUMBER OF TUBES PER VERTICAL ROW OF
C      THE CONDENSER.
C      TSAT=SATURATION TEMPERATURE.
C      TW=TEMPERATURE OF THE WALL SURFACE OF THE TUBES.
C      ADG=ACCELERATION DUE TO GRAVITY.
C      CONG=CONSTANT TO BE APPLIED TO THE FUNCTION FC
C      WHICH CONTRIBUTES TO THE GRAVITY EFFECT ON THE
C      CONDENSATE DRAIN-OUT RATE.
C      CONST=CONSTANT TO BE APPLIED TO THE FUNCTION FB
C      WHICH CONTRIBUTES TO THE SURFACE-TENSION EFFECT
C      ON THE CONDENSATE DRAIN-OUT RATE.
C      X=ANGLE IN RADIANS, MEASURED FROM THE UPPER
C      GENERATRIX OF THE POLAR PROFILE.
C      NO=No. OF OBSERVATIONS.
C      IN=VALUE OF THE INTEGRATION.
C      H=STEP SIZE,USED IN EVALUATING THE INTEGRAL.
C      HTC=HEAT TRANSFER COEFFICIENT.
C      DELTA=BOUNDARY LAYER THICKNESS OF THE CONDENSATE
C      FORMED OVER THE TUBE-SURFACE.
C      NOTE:-- ALL VARIABLES ARE MEASURED IN 'SI' UNITS.
C      ******************************************************
       REAL F,IO,IN,KL,MUL
       DIMENSION Y(100),HTC(100)
       FA(X)=X/X
       FB(X)=0.*(CONST)*(0.5)*(EXP(-2.*X))
       FC(X)=(CONG)*(COS(X))
       FD(X)=FB(X)+FC(X)
       FE(X)=(FD(X))**(1./3.)
       F(X)=FE(X)*FA(X)
```

```
      WRITE(22,3)
      FORMAT(///////////////)
      RMF=1.0
      RAD=.011
      A=(RAD)*(RMF)
      RHOL=988.1638
      FL=0.654157
      MUL=(5.180925)/(10.**4)
      SIGL=.074
      HFG=2364064.0
      HFGSL=2369531.0
      CCFSC=1.048099
      ANT=40.0
      TSAT=57.76
      TW=54.28
      DT=TSAT-TW
      ADG=9.81
      CONG1=(RHOL**2)*(HFGSC)*(ADG)
      CONG2=(3.)*(FL)*(MUL)*(DT)*(A)
      CONG=CONG1/CONG2
      CONST1=(SIGL)*(RHOL)*(HFGSC)
      CONST2=(3.)*(FL)*(A**3)*(MUL)*(DT)
      CONST=CONST1/CONST2
C     ************************************************************
C     EVALUATION OF THE INTEGRAL 'IN' BY USING SIMPSON'S
C     RULE FOR NUMERICAL INTEGRATION.
C     ************************************************************
      X1=1.5759329
      X2=1.6021222
      NO=0
   99 H=(X2-X1)...
      I=2
      S1=F(X1)+F(X2)
      S2=0.
      S4=F(X1+H)
      IO=0.
      IN=(S1+4.*S4)*(H/3.)
   28 IF(ABS((IN-IO)/IN).LE..001)GO TO 29
      S2=S2+S4
      S4=0.
      X=X1+H/2.
      DO 10 J=1,I
      S4=S4+F(X)
      X=X+H
   10 CONTINUE
      H=H/2.
      I=2*I
      IO=IN
      IN=(S1+2.*S2+4.*S4)*(H/3.)
      GO TO 28
   29 DELTA=(((IN)**(0.25))/FE(X2))*(1.0745699)
      HTC(I)=(KL/DELTA)*(CCFSC)/(ANT**0.25)
      Y(I)=HTC(I)
      WRITE(24,*)Y(I)
      NO=NO+1
      WRITE(22,1)NO,X2,HTC(I)
    1 FORMAT(/20X,I2,5X,F9.7,5X,F9.3)
```

```
10300   C      *******************************************************
10400   C      THE FOLLOWING FILE STORES THE SOLUTION,GIVING THE
10500   C      HEAT TRANSFER COEFFICIENTS AT DIFFERENT DISCRETE
10600   C      LOCATIONS,SPECIFIED BY 'X2' AND THE TOTAL No. OF
10700   C      SUCH REPITITIVE CALCULATIONS ARE PERFORMED BY THE
10800   C      THE 'IF' LOOP THAT FOLLOWS THE 'WRITE' STATEMENT.
10900   C      *******************************************************
11000          IF(X2.GT.4.680970)GO TO 59
11100          X2=X2+0.0314159
11200          GO TO 39
11300   59     CLOSE(UNIT=24,DEVICE='DSK')
11400          READ(24,*)(Y(I),I=1,99)
11500   C      **********************************************************
11600   C      THE FOLLOWING FILE STORES THE HEAT TRANSFER COEFFICIENT
11700   C      DATA,TO BE USED IN THE PROGRAMME FOR FITTING A POLYNOMI-
11800   C      AL THROUGH THESE DATA POINTS.
11900   C      **********************************************************
12000          WRITE(24,*)(Y(I),I=1,99)
12100          STOP
12200          END
12300   C      ***************************************************************
P
12400   C      NESSARY CHANGES,WHICH ARE TO BE MADE IN THE ABOVE PROGRAMME
12500   C      TO USE THE SAME FOR SOME OTHER POLAR PROFILES,ARE AS
12600   C      FOLLOWS:--
12700   C      (a): FOR THE PROFILE 'R=A*THETA':--------
12800   C         1.  THE EXPRESSION FOR 'RMF' IN LINE No. '4900' IS
12900   C      CHANGED TO 'RMF=0.3021'.
13000   C         2.THE EXPRESSION FOR 'FA(X)' IN LINE No. '4300' IS
13100   C      CHANGED TO 'FA(X)=SQRT(1.0+X**2)'.
13200   C         3.THE EXPRESSION FOR 'FB(X)' IN LINE No. '4400' IS
13300   C      CHANGED TO 'FB(X)=(CONST)*((X**3+4.*X)/((1.+X**2)**3)) .
13400   C      (b): FOR THE PROFILE ,R=A*EXP(THETA)':--------
13500   C         1.THE EXPRESSION FOR 'RMF' IN LINE No. '4900' IS
13600   C      CHANGED TO 'RMF=0.02085'.
13700   C         2.THE EXPRESSION FOR 'FA(X)' IN LINE No. '4300' IS
13800   C      CHANGED TO 'FA(X)=SQRT(2.)*EXP(X)'.
13900   C         3.THE EXPRESSION FOR 'FB(X)' IN LINE No. '4400' IS
14000   C      CHANGED TO 'FB(X)=(CONST)*(0.5)*(EXP(-2.*X))'.
14100   C               TO STUDY THE ISOLATED EFFECT OF GRAVITY ON
14200   C      THE HEAT TRANSFER COEFFICIENT AUGMENTATION,IN BOTH
14300   C      THE ABOVE CASES '(a)' AND 'b',THE EXPRESSION FOR
14400   C      FB(X) IS MULTIPLIED BY '0.0' TO NULLIFY THE CON-
14500   C      TRIBUTION OF SURFACE-TENSION.
14510   C      ***********************************************************

.type y
```

APPENDIX-7

```
C                  -----------------------------
C                  COMPUTER PROGRAMME No.4
C                  -----------------------------
C     ******************************************************
C     PROGRAMME TO DETERMINE THE FUNCTIONAL RELATION
C     BETWEEN THE STEAM-SIDE HEAT TRANSFER COEFFICIE-
C     NT 'HS' AND ANGLE 'THETA',MEASURED FROM THE UP-
C     PER GENERATRIX,USING THE 'METHOD OF LEAST SQUARES .
C     ******************************************************
      DIMENSION S(10,10),X(100),B(5),A(5),Y(100),CY(100),PD(100)
C     ******************************************************
C     NOMENCLATURE: --
C                  M=No. OF DATA POINTS.
C                  X='THETA',MEASURED IN RADIAN.
C                  Y='HS',ACTUALLY OBTAINED.
C                  N=ORDER OF THE POLYNOMIAL TO BE FITTED.
C                  CY='HS',CALCULATED FROM THE POLYNOMIAL
C                  RELATION.
C                  PCD=PERCENTAGE DEVIATION BETWEEN 'Y' AND
C                  'CY'.
C                  SUM=CUMULATIVE 'PCD'.
C     ******************************************************
      M=99
      X(1)=1.6022122
      DO 2 J=1,M
      X(J)=X(1)+(J-1)*(0.0314159)
    2 CONTINUE
      N=3
C     ******************************************************
C     THE FOLLOWING FILE READS THE VALUES OF 'HS',ALREADY
C     DETERMINED FROM ANOTHER PROGRAMME.
C     ******************************************************
      READ(23,*)(Y(I),I=1,M)
      SUMY=0.
      SUMXY=0.
      SUMXSY=0.
      SUMXCY=0.
C     SUMXFY=0.
      SUMX=0.
      SUMXSR=0.
      SUMXCB=0.
      SUMXFR=0.
      SUMXFV=0.
      SUMXSX=0.
C     SUMXSN=0.
C     SUMXET=0.
      DO 10 I=1,M
      SUMY=SUMY+Y(I)
      SUMXY=SUMXY+X(I)*Y(I)
      SUMXSY=SUMXSY+(X(I)**2)*Y(I)
      SUMXCY=SUMXCY+(X(I)**3)*Y(I)
C     SUMXFY=SUMXFY+(X(I)**4)*Y(I)
      SUMX=SUMX+X(I)
```

```
      SUMXSR=SUMXSR+X(I)**2
      SUMXCB=SUMXCB+X(I)**3
      SUMXFR=SUMXFR+X(I)**4
      SUMXFV=SUMXFV+X(I)**5
      SUMXSX=SUMXSX+X(I)**6
C     SUMXSN=SUMXSN+X(I)**7
C     SUMXET=SUMXET+X(I)**8
10    CONTINUE
C     *******************************************************
C     'S' REPRESENTS THE COEFFICIENT MATRIX,GENERATED FROM
C     THE SIMULTANEOUS LINEAR EQUATIONS,OBTAINED FOR SOLVI-
C     NG THE LEAST SQUARE METHOD.
C     *******************************************************
      S(1,1)=M
      S(1,2)=SUMX
      S(1,3)=SUMXSR
      S(1,4)=SUMXCB
C     S(1,5)=SUMXFR
      S(2,1)=SUMX
      S(2,2)=SUMXSR
      S(2,3)=SUMXCB
      S(2,4)=SUMXFR
C     S(2,5)=SUMXFV
      S(3,1)=SUMXSR
      S(3,2)=SUMXCB
      S(3,3)=SUMXFR
      S(3,4)=SUMXFV
C     S(3,5)=SUMXSX
      S(4,1)=SUMXCB
      S(4,2)=SUMXFR
      S(4,3)=SUMXFV
      S(4,4)=SUMXSX
C     S(4,5)=SUMXSN
C     S(5,1)=SUMXFR
C     S(5,2)=SUMXFV
C     S(5,3)=SUMXSX
C     S(5,4)=SUMXSN
C     S(5,5)=SUMXET
      B(1)=SUMY
      B(2)=SUMXY
      B(3)=SUMXSY
      B(4)=SUMXCY
C     B(5)=SUMXFY
      N=N+1
C     *****************************************************
C     THE FOLLOWING FILE STORES THE ELEMENTS OF THE
C     COEFFICIENT MATRIX 'S'.
C     *****************************************************
      WRITE(21,*)((S(I,J),J=1,N),I=1,N),(B(I),I=1,N)
      CLOSE(UNIT=21,DEVICE='DSK')
      READ(21,*)((S(I,J),J=1,N),I=1,N),(B(I),I=1,N)
C     ****************************
C     INVERSION OF MATRIX 'S'.
C     ****************************
      I=1
      NX=N+1
      NY=2*N
      DO 80 J=NX,NY
      S(I,J)=1.
      I=I+1
80    CONTINUE
      L=1
      K=2
110   XM=S(L,L)
```

```
      DO 140 J=L,NY
      S(L,J)=S(L,J)/XM
 140  CONTINUE
      DO 190 I=K,N
      X1=S(I,L)
      DO 191 J=L,NY
      S(I,J)=S(I,J)-S(L,J)*X1
 191  CONTINUE
 190  CONTINUE
      L=L+1
      K=K+1
      IF(L-N)110,110,230
 230  L=N
 235  LZ=L-1
      DO 290 K=1,LZ
      I=L-K
      Y1=S(I,L)
      DO 291 J=L,NY
      S(I,J)=S(I,J)-S(L,J)*Y1
 291  CONTINUE
 290  CONTINUE
      L=L-1
      IF(L-1)320,320,235
 320  WRITE(43,*)((S(I,J),J=NX,NY),I=1,N)
      CLOSE(UNIT=43,DEVICE='DSK')
      READ(43,*)((S(I,J),J=NX,NY),I=1,N)
C     ************************************************************
C     MULTIPLICATION OF INVERTED MATRIX OF 'S' WITH MATRIX 'B'.
C     ************************************************************
      DO 70 I=1,N
      A(I)=0.
      DO 71 J=NX,NY
      A(I)=A(I)+S(I,J)*B(J-N)
 71   CONTINUE
 70   CONTINUE
      WRITE(44,*)(A(I),I=1,(NY-N))
      CLOSE(UNIT=44,DEVICE='DSK')
C     ******************************************************
C     MATRIX 'A' STORES THE VALUES OF THE COEFFICIENTS OF THE
C     POLYNOMIAL EXPRESSION.
C     ******************************************************
      READ(44,*)(A(I),I=1,(NY-N))
C     ***************************************************
C     EVALUATION OF THE PERCENTAGE DIFFERENCE BETWEEN THE
C     ACTUAL VALUE OF 'HS'  AND THE VALUE,CALCULATED FROM
C     THE POLYNOMIAL,FITTED THROUGH THE ACTUAL  VALUES.
C     ***************************************************
      SUM=0.
      DO 27 J=1,M
      CY(J)=0.
      DO 28 I=1,N
      CY(J)=CY(J)+(A(I)*(X(J)**(I-1)))
 28   CONTINUE
      PD(J)=((CY(J)-Y(J))/Y(J))*100.
      SUM=SUM+PD(J)
      WRITE(45,*)X(J),Y(J),CY(J),PD(J),SUM
 27   CONTINUE
      STOP
      END
```

.type w

APPENDIX-8

COMPUTER PROGRAMME No.5

```
C        ********************************************************
C        PROGRAMME TO PLOT THE POLAR PROFILE OF THE CONDENSER
C        TUBES BY USING 'INTERACTIVE GRAPHICS LIBRARY(IGL)'
C        SUBROUTINES(FOR PROFILE 'R=A*THETA').
C        ********************************************************
C        MAIN PROGRAMME TO DRAW THE P.I.R.C. PROFILE.
C        NOMENCLATURE:-----
C        RAD=RADIUS OF THE CIRCULAR PROFILE,THAT HAS BEEN
C        MOULDED INTO THE POLAR PROFILE OF THE FORM
C        'R=A*THETA' OR 'R=A*EXP(THETA)'.
C        THETA=ANGLE,MEASURED FROM THE UPPER GENARATRIX.
C        P.I.R.C.=PROGRESSIVELY INCREASING RADIUS OF CURVATURE.
C        ********************************************************
         DIMENSION R(25),THETA(25)
C        INITIALISATION OF THE GRAPHICS TERMINAL.
         CALL GRSTRT(4010,5)
         CALL MILLIM
         CALL VWPORT(0.,100.,0.,100.)
         CALL WINDOW(0.,100.,0.,100.)
         THETA(1)=90.
         RAD=11.0
         A=0.3021*RAD
C        A=0.02085*RAD
         R(1)=A*(3.14159/2.)
C        R(1)=A*EXP(3.14159/2.)
         DO 9 J=2,21
         THETA(J)=THETA(1)+(J-1)*(180./20.)
         R(J)=(A)*((3.14159/2.)+(J-1)*(3.14159/20.))
C        R(J)=(A)*EXP((3.14159/2.)+(J-1)*(3.14159/20.))
9        CONTINUE
C        ********************************************************
C        THE FOLLOWING FILE  STORES THE SETS OF DATA POINTS
C        TO DRAW THE P.I.R.C. PROFILE(LEFT HALF SIDE).
C        ********************************************************
         WRITE(23,*)(R(I),THETA(I),I=1,21)
         CLOSE(UNIT=23,DEVICE='DSK')
         READ(23,*)(R(I),THETA(I),I=1,21)
C        ********************************************************
C        THE FOLLOWING 'CALL' STATEMENT CALLS THE SUBROUTINE
C        'FIGURE' TO DRAW THE P.I.R.C. PROFILE(LEFT HALF SIDE).
C        ********************************************************
         CALL FIGURE(R,THETA)
         DO 99 J=2,21
         THETA(J)=360.+180.-(THETA(1)+(J-1)*(180./20.))
         R(J)=A*((3.14159/2.)+(J-1)*(3.14159/20.))
C        R(J)=(A)*EXP((3.14159/2.)+(J 1)*(3.14159/20.))
99       CONTINUE
C        ********************************************************
C        THE FOLLOWING FILE STORES THE DATA POINTS TO DRAW
C        THE P.I.R.C. PROFILE(RIGHT HALF SIDE).
C        ********************************************************
         WRITE(30,*)(R(J),THETA(J),J=1,21)
         CLOSE(UNIT=30,DEVICE='DSK')
         READ(30,*)(R(J),THETA(J),J=1,21)
C        ********************************************************
C        THE FOLLOWING CALL TO SUBROUTINE 'FIGURE' DRAWS THE
C        P.I.R.C. PROFILE(RIGHT HALF SIDE).
```

```
05800   C      ************************************************************
05900          CALL FIGURE(R,THETA)
06000   C      *******************************
06100   C      DRAWING THE CIRCULAR PROFILE.
06200   C      *******************************
06300          CALL MOVE(80.0,80.0)
06400          CALL ARC(RAD,0.0,360.0)
06500          CALL GRSTOP
06600          END
06700   C      *****************************************************
06800   C      THE FOLLOWING SUBROUTINE 'FIGURE' IS USED IN THE
06900   C      *****************************************************
07000          SUBROUTINE FIGURE(R,THETA)
07100          DIMENSION R(25),THETA(25)
07200          CALL PIVOT(50.0,85.0)
07300          CALL MPOLAR(R(1),THETA(1))
07400          DO 90 J=2,21
07500          CALL SMOOTH
07600          CALL GRAIN(0.0)
07700          CALL DPOLAR(R(J),THETA(J))
07800    90    CONTINUE
07900          RETURN
08000          END
08005   C      *****************************************************
08010   C      THE FOLLOWING CHANGES ARE TO BE MADE TO DRAW THE
08110   C      POLAR PROFILE 'R=A*EXP(THETA)' BY USING THE SAME
08210   C      PROGRAMME.
08310   C              DETACH THE LABEL 'C', THAT IS, THE COMMENT
08410   C      STATEMENT FROM LINE No.S  '2300','2500','2900',
08510   C      '4600' AND ATTACH THE LABEL 'C' TO THE LINES,NUM-
08610   C      BERED '2200','2400','2800'AND '4500'.
08615   C      *****************************************************

.ty\yt\^C

.type za
```

APPENDIX-9

COMPUTER PROGRAMME No.6

```
C        ******************************************************
C        PROGRAMME FOR DRAWING THE GRAPH OF HEAT TRANSFER
C        COEFF.VS.THETA FOR POLAR PROFILE TUBES,WHEN A VA-
C        POUR IS CONDENSING OVER THE TUBES,USING THE INTERA-
C        CTIVE GRAPHICS LIBRARY(IGL) SUBROUTINES.
C        ******************************************************
C        NOMENCLATURE:--
C        X=ANGLE THETA,MEASURED IN RADIANS FROM THE UPPER
C        GENARATRIX OF THE POLAR CURVE.
C        Y=HEAT TRANSFER COEFFICIENT.
C        P.I.R.C.=PROGRESSIVELY INCREASING RADIUS OF CURVATURE.
C        ******************************************************
         DIMENSION X(300),Y(300),AX(100),AY(100)
         X(1)=1.6022122
         DO 98 I=2,99
         X(I)=X(1)+.0314159*(I-1)
98       CONTINUE
C        ******************************************************
C        THE FOLLOWING FILE READS THE HEAT TRANSFER COEFF.
C        DATA FOR CIRCULAR PROFILE TUBES FROM ANOTHER PRO
C        GRAMME,WHERE IT IS CALCULATED AND STORED IN THE
C        SAME FILE.
C        ******************************************************
         READ(20,*)(Y(I),I=1,99)
C        *******************************************
C        INITIALISATION OF THE GRAPHICS TERMINAL.
C        *******************************************
         CALL GRSTRT(4010,5)
         CALL VWPORT(10.0,110.0,0.0,100.0)
         CALL WINDOW(-1.,5.5,-1000.,14500.)
         XMIN=0.
         XMAX=5.
         YMIN=0.
         YMAX=14000.
         CALL MOVE(XMIN,YMIN)
         CALL DRAW(XMIN,YMAX)
         CALL MOVE(XMIN,YMIN)
         CALL DRAW(XMAX,YMIN)
C        **********************
C        LABELLING THE X-AXIS
C        **********************
         CALL TXICUR(8)
         DO 100 IX=1,6,1
         AX(IX)=FLOAT(IX-1)
         CALL MOVE(AX(IX),0.0)
         CALL DRAW(AX(IX),-500.0)
         CALL MOVE(AX(IX),-650.0)
         CALL RNUMBR(AX(IX),1,3)
100      CONTINUE
C        **********************
C        LABELLING THE Y-AXIS
C        **********************
         CALL TXICUR(1)
         DO 200 IY=1,14001,1000
         AY(IY)=FLOAT(IY-1)
         CALL MOVE(0.0,AY(IY))
         CALL DRAW(-0.1,AY(IY))
         CALL MOVE(-0.9,AY(IY))
```

```
              CALL RNUMBR(AY(IY),1,7)
  200         CONTINUE
C             ***********************************************
C             DRAWING THE GRAPH FOR CIRCULAR PROFILE TUBES.
C             ***********************************************
              CALL MOVE(X(1),Y(1))
              DO 10 J=2,99
              CALL SMOOTH
              CALL GRAIN(0.0)
              CALL DRAW(X(J),Y(J))
  10          CONTINUE
C             *********************************************************
C             THE FOLLOWING FILE READS THE HEAT TRANSFER COEFF.
C             DATA FROM ANOTHER PROGRAMME,WHERE IT IS CALCULATED
C             AND STORED IN THE SAME FILE.(FOR P.I.R.C. PROFILE TUBES
C             WITH BOTH  GRAVITY AND SURFACE-TENSION EFFECTS.
C             *********************************************************
              READ(23,*)(Y(I),I=1,99)
C             ***********************************************************
C             DRAWING THE GRAPH FOR P.I.R.C. PROFILE CONSIDERING
C             GRAVITY AND SURFACE-TENSION SIMULTANEOUSLY.
C             ***********************************************************
              CALL MOVE(X(1),Y(1))
              DO 20 K=2,99
              CALL SMOOTH
              CALL GRAIN(0.0)
              CALL DRAW(X(K),Y(K))
  20          CONTINUE
C             ***********************************************************
C             THE FOLLOWING FILE READS THE HEAT TRANSFER COEFF. DATA
C             FROM ANOTHER PROGRAMME,WHERE IT IS CALCULATED AND STOR
C             ED IN THE SAME FILE.(FOR P.I.R.C. PROFILE TUBES WITH
C             GRAVITY EFFECT ONLY)
C             ***********************************************************
              READ(24,*)(Y(I),I=1,99)
C             ***********************************************************
C             DRAWING THE GRAPH FOR P.I.R.C. PROFILE CONSIDERING
C             GRAVITY ONLY.
C             ***********************************************************
              CALL MOVE (X(1),Y(1))
              DO 30 M=2,99
              CALL SMOOTH
              CALL GRAIN(0.0)
              CALL DRAW (X(M),Y(M))
  30          CONTINUE
              CALL GRSTOP
              END

.^C

.^C

.type u
```

APPENDIX -10

COST OF PRODUCTION OF ELECTRICAL ENERGY
(SOURCE: PANKI THERMAL POWER STATION, KANPUR)

Yr. of observation	Year	Energy cost/unit [Paise]
1.	1979-80	32.50
2.	1980-81	35.67
3.	1981-82	40.96
4.	1982-83	46.24
5.	1983-84	46.24
6.	1984-85	57.50
7.	1985-86	72.55

APPENDIX-11

```
C                   ---------------------------
C                   COMPUTER PROGRAMME No.7
C                   ---------------------------
C     ************************************************************
C     PROGRAMME TO EVALUATE THE FUNCTIONAL RELATIONSHIP
C     OF'CEPU',COST OF PRODUCTION OF ENERGY WITH'TIME'
C     BY EMPLOYING THE'METHOD OF LEAST SQUARES'.
C     ************************************************************
      DIMENSION S(10,10),X(100),B(5),A(5),Y(100),CY(100),PD(100)
C     ************************************************************
C     NOMENCLATURE:---
C     M=NUMBER OF DATA POINTS
C     N=ORDER OF POLYNOMIAL TO BE FITTED THROUGH THE DATA POINTS
C     X=INDEPENDENT VARIABLE(TIME IN 'YEAR',REPRESENTED BY
C     THE No. OF OBSERVATION.)
C     Y=DEPENDENT VARIABLE('CEPU')
C     CEPU=COST OF PRODUCTION OF ENERGY.
C     CY=DEPENDENT VARIABLE CALCULATED FROM THE POLYNOMIAL-FIT.
C     PD=PERCENTAGE DEVIATION
C     SUM=CUMULATIVE 'PD'.
C     ************************************************************
      WRITE(49,31)
   31 FORMAT(/////////////////)
      M=7
      N=2
C     ************************************************************
C     THE FOLLOWING FILE GIVES INPUT DATA TO THE PROGRAMME
C     ************************************************************
      READ(24.*)(X(I),Y(I),I=1,M)
C     ************************************************************
C     EVALUATION OF THE ELEMENTS OF THE MATRIX 'S' AND MATRIX 'B'.
C     ************************************************************
      SUMY=0.
      SUMXY=0.
      SUMXSY=0.
      SUMXCY=0.
      SUMX=0.
      SUMXSR=0.
      SUMXCB=0.
      SUMXFR-0.
      SUMXFV=0.
      SUMXSX=0.
      DO 10 I=1,M
      SUMY=SUMY+Y(I)
      SUMXY=SUMXY+X(I)*Y(I)
      SUMXSY=SUMXSY+(X(I)**2)*Y(I)
      SUMXCY=SUMXCY+(X(I)**3)*Y(I)
      SUMX=SUMX+X(I)
      SUMXSR=SUMXSR+X(I)**2
      SUMXCB=SUMXCB+X(I)**3
      SUMXFR=SUMXFR+X(I)**4
      SUMXFV=SUMXFV+X(I)**5
      SUMXSX=SUMXSY+X(I)**6
   10 CONTINUE
      S(1,1)=M
```

```
                S(1,2)=SUMX
                S(1,3)=SUMXSR
      C         S(1,4)=SUMXCB
                S(2,1)=SUMX
                S(2,2)=SUMXSR
                S(2,3)=SUMXCB
      C         S(2,4)=SUMXFR
                S(3,1)=SUMXSR
                S(3,2)=SUMXCB
                S(3,3)=SUMXFR
      C         S(3,4)=SUMXFV
      C         S(4,1)=SUMXCB
      C         S(4,2)=SUMXFR
      C         S(4,3)=SUMXFV
      C         S(4,4)=SUMXSX
                B(1)=SUMY
                B(2)=SUMXY
                B(3)=SUMXSY
      C         B(4)=SUMXCY
                N=N+1
                WRITE(21,*)((S(I,J),J=1,N),I=1,N),(B(I),I=1,N)
                CLOSE(UNIT=21,DEVICE='DSK')
                READ(21,*)((S(I,J),J=1,N),I=1,N),(B(I),I=1,N)
      C         *******************************
      C         INVERSION OF THE MATRIX 'S'.
      C         *******************************
                I=1
                NX=N+1
                NY=2*N
                DO 80 J=NX,NY
                S(I,J)=1.
                I=I+1
      80        CONTINUE
                L=1
                K=2
      110       XM=S(L,L)
                DO 140 J=L,NY
                S(L,J)=S(L,J)/XM
      140       CONTINUE
                DO 190 I=K,N
                X1=S(I,L)
                DO 191 J=L,NY
                S(I,J)=S(I,J)-S(L,J)*X1
      191       CONTINUE
      190       CONTINUE
                L=L+1
                K=K+1
                IF(L-N)110,110,230
      230       L=N
      235       LZ=L-1
                DO 290 K=1,LZ
                I=L-K
                Y1=S(I,L)
                DO 291 J=L,NY
                S(I,J)=S(I,J)-S(L,J)*Y1
      291       CONTINUE
      290       CONTINUE
                L=L-1
                IF(L-1)320,320,[illegible]
      320       WRITE(23,*)((S(I,J),J=NX,NY),I=1,N)
                CLOSE(UNIT=23,DEVICE='DSK')
                READ(23,*)((S(I,J),J=NX,NY),I=1,N)
```

```
C     ****************************************************************
C     MULTIPLICATION OF THE INVERTED MATRIX OF 'S' WITH MATRIX 'P'.
C     ****************************************************************
      DO 70 I=1,N
      A(I)=0.
      DO 71 J=NX,NY
      A(I)=A(I)+S(I,J)*P(J)
   71 CONTINUE
   70 CONTINUE
C     ****************************************************************
C     THE ROW MATRIX 'A' GIVES THE COEFF.S OF THE FITTED POLYNOMIAL.
C     ****************************************************************
      WRITE(48,*)(A(I),I=1,(NY-N))
      CLOSE(UNIT=48,DEVICE='DSK')
      READ(48,*)(A(I),I=1,(NY-N))
C     ****************************************************************
C     CALCULATION OF THE DEVIATION BETWEEN THE ACTUAL VALUE AND
C     CALCULATED VALUE OF THE DEPENDENT VARIABLE.
C     ****************************************************************
      SUM=0.
      DO 27 J=1,M
      CY(J)=0.
      DO 28 I=1,N
      CY(J)=CY(J)+(A(I)*(X(J)**(I-1)))
   28 CONTINUE
      PD(J)=((CY(J)-Y(J))/Y(J))*100.
      SUM=SUM+PD(J)
      WRITE(49,*)X(J),Y(J),CY(J),PD(J),SUM
   27 CONTINUE
      STOP
      END
```

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APPENDIX -12

AVERAGE ROUGHNESS OF COMMERCIAL PIPES [6]

Material	Absolute roughness of pipe (in mm.)
Riveted steel	0.9 -9.0
Concrete	0.3 - 3.0
Wood stave	0.18- 0.9
Cast Iron	0.26
Galvanized Iron	0.15
Asphalted Cast Iron	0.12
Commercial Steel or Wrought Iron	0.046
Drawn Tubing	0.0015
Glass	Smooth

APPENDIX-13

```
C              ------------------------------
C              COMPUTER PROGRAMME No.8
C              ------------------------------
C      ****************************************************
C      PROGRAMME FOR OBTAINING THE OPTIMUM CONDENSER
C      TUBE DIMENSIONS ON THE BASIS OF MINIMUM TOTAL
C      COST FOR CIRCULAR TUBE PROFILE.
C      ****************************************************
C              NOMENCLATURE----
C              --------------------
C      AF=AREA FACTOR
C      MUL=DYNAMIC VISCOSITY OF  LIQUID CONDENSATE
C      KL=THERMAL CONDUCTIVITY OF LIQUID CONDENSATE
C      LT=LENGTH OF TUBES
C      CEPU=COST OF PRODUCTION  OF ENERGY
C      HTCF=HEAT TRANSFER COEFFICIENT FACTOR
C      QOR=ORIGINAL HEAT TRANSFER RATE IN THE CONDANSER
C      ETAP=OVERALL EFFICIENCY OF THE PUMP
C      DO=OUTER DIAMETER OF TUBES
C      DI=INNER DIAMETER OF TUBES
C      EPSN=ABSOLUTE ROUGHNESS OF TUBES
C      INTRT=RATE OF INTEREST
C      RHOL=MASS-DENSITY OF THE LIQUID CONDENSATE
C      LMTD=LOG MEAN TEMPERATURE DIFFERENCE
C      PR=PRANDTL No.
C      HS=STEAM-SIDE HEAT TRANSFER COEFFICIENT
C      RHOM=MASS-DENSITY OF MATERIAL,USED IN THE TUBES
C      CM=COST OF MATERIAL
C      VFR=VOLUME FLOW RATE OF COOLING WATER IN THE CONDENSER
C      PER= PERIMETER OF TUBES
C      ACN=AREA OF CROSS SECTION
C      RR=RELATIVE ROUGHNESS OF TUBES
C      VEL=VELOCITY OF COOLING WATER
C      HMD=HYDRAULIC MEAN DIAMETER
C      REN=REYNOLD'S No.
C      FF=FRICTION FACTOR
C      HW=WATER-SIDE HEAT TRANSFER COEFFICIENT
C      THRS=THERMAL RESISTANCE OF STEAM SIDE
C      THRW=THERMAL RESISTANCE OF WATER SIDE
C      THRM=MISCELLENEOUS THERMAL RESISTANCE
C      UO=OVERALL HEAT TRANSFER COEFFICIENT,BASED
C      ON OUTER SURFACE AREA
C      AO=OUTER SURFACE AREA
C      DP=PRESSURE DROP OF COOLING WATER IN CONDENSR TUBES
C      DYNP=DYNAMIC PRESSURE OF COOLING WATER
C      PWRT=THEORITICAL PUMPING POWER
C      PWRP=ACTUAL PUMPING  POWER
C      ENCY=ENERGY CONSUMPTION PER YEAR
C      CEN=COST OF ENERGY
C      CENT=TOTAL COST OF ENERGY
C      PWENC=PRESENT WORTH OF ENERGY COST
C      VOLM=VOLUME OF MATERIAL,USED
```

```
          UO=1./SUMTHR
          AO=(QOR)/(UO*LMID)
          LT=(AO)/(PI*DON(I))
          DP=(FF)*(LT)*(VEL**2)*(RHOL)/(2.0*DIN)
          DYNP=(0.5)*(RHOL)*(VEL**2)
          DP=DYNP+DP
          PWRT=(DP)*(VFR)
          PWRP=(PWRT)/(ETAP)
          ENCY=(PWRP)*(24.0)*(365.0)/(1000.0)
          CENT=0.0
          DO 9 J=8,32
          X=FLOAT(J)
          CEN=ENCY*CEPU(X)/100.0
          PWENC=CEN/((1.0+INTRT)**(J-8))
          CENT=CENT+PWENC
9         CONTINUE
          VOLM=(3.14159/4.)*((DOUT(I)**2)-(DIN**2))*(LT)
          MTM=(VOLM)*(RHOM)
          CMT=(MTM)*(CM)
          CT(I)=CENT+CMT
C         ************************************************************
C         THE FOLLOWING FILE STORES THE RESULTS OF THE ABOVE ANALYSIS.
C         ************************************************************
          WRITE(47,7)DON(I),FF,ENCY,CENT,CMT,CT(I),MTM,LT
7         FORMAT(/5X,F5.3,3X,F10.8,3X,F15.8,3X,F11.2,3X,F8.2,
          13X,F11.2,3X,F7.3,3X,F6.2)
10        CONTINUE
          STOP
          END
C         *****************************************************
C         SUBROUTINE FOR OBTAINING FRICTION FACTOR FROM
C         THE IMPLICIT RELATION BY 'COLEBROOK' USING NE-
C         WTON-RAPHSON TECHNIQUE.
C         *****************************************************
          SUBROUTINE FRIXON(RR,REN,FF)
          X=0.005
19        DFX1=2.51/(REN*SQRT(X))
          DFX2=RR/3.7
          DFX3=DFX1+DFX2
          FX1=(+0.86859)*(ALOG(DFX3))
          FX2=1./SQRT(X)
          FX=FX1+FX2
          DFXA=(1./DFX3)*(0.86859)
          DFXB=(-1.255/REN)/(X**1.5)
          DFXC=(-0.5)/(X**1.5)
          DFX=(DFXA*DFXB)+(DFXC)
          XN=(X)-(FX/DFX)
          IF((ABS(XN-X)).LE.0.000001)GO TO 29
          X=XN
          GO TO 19
29        FF=XN
          RETURN
          END
C         ******************************************************
C         NOTE:--
C                      TO USE THE SAME PROGRAMME FOR POLAR PROFILES
C         OTHER THAN CIRCULAR,THE VALUES OF 'AP' AND 'HTCF'
C ARE TO BE CHANGED SUITABLY,AS DISCUSSED IN THE
C         'CHAPTER.2' OF THE THESIS.HOWEVER,THE CORRESPONDING
C         VALUES OF THE ABOVE PARAMETERS ARE FIXED AS 'UNITY'
C         FOR CIRCULAR PROFILE TUBES
C         ******************************************************
```

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